A COMPRESSOR TEST FACILITY

GUSTAV F. SWAINSON, JR.
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1951

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Thesis
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ABSTRACT

Title: A Compressor Test Facility

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Submitted for the degrees of Naval Engineer and Master of Science in the Department of Naval Architecture and Marine Engineering on 18 May 1951.

The operation of gas turbine units over a long period of time had previously been restricted by failure of metals in service. However, with the increasing use of gas turbines on land and sea installations, it is necessary to know at what point the units must be torn down for overhaul. The design of turbines and combustion chambers are relatively insensitive to changes in efficiency due to fouling. The compressor, however, is quite sensitive and small changes in blade shapes effect large changes in efficiency. For this reason it is necessary to study the effect of fouling on compressor blading. This fouling can come from several sources -- salt particles in the atmosphere over the sea, or dust particles over land.

In order to study the effect of this fouling on a compressor, it was necessary that a compressor test facility be designed and built, and this thesis concerned itself with this project.

A Westinghouse X9.5B jet engine was used as the machinery element of this test facility. However, since it was not desired to run the apparatus "hot", a change in the air flow had to be made. A power air circuit including the turbine wheel comprised the driving unit for the apparatus, and a test air

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circuit including the compressor made up the experimental circuit.

In order to accomplish the flow of two circuits through the gas turbine, the combustion chamber was stripped of all its burner elements and a diaphragm was inserted transversly inside the chamber. An annulus was mounted on one side of the diaphragm to accommodate the flow of power air and an exhaust duct was tapped into the other side of the diaphragm to receive the flow from the compressor outlet.

An oil mist recovery system was designed, built, and installed in the apparatus in order to prevent the fouling of the wind tunnel ducting with exhaust lubricant.

Measurement of the air flow through the compressor is accomplished by measuring the pressure drop across the inlet duct which has been calibrated against a standard orifice.

Test runs were made with the apparatus at speeds up to 15,000 rpm in order to determine any mechanical difficulties and data obtained during these runs gave an approximation to the compressor characteristic curves at speeds of 15000 rpm and below.

Cambridge, Massachusetts
May 18, 1951

Professor J.S. Newell
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge, Massachusetts

Dear Sir:

In accordance with the requirements for the Degrees of Naval Engineer and Master of Science, We submit herewith a thesis entitled " A Compressor Test Facility."

Respectfully,

ACKNOWLEDGEMENTS

The authors wish to express their thanks to Professor E. S. Taylor, Associate Professor E. P. Neumann, and Research Associate F. Lustwerk for their invaluable assistance in developing design features for the thesis.

The authors are greatly indebted to Mr. Lustwerk for his valuable assistance in the laboratory, particularly during the calibration run and the operating run.

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CHAPTER I - PURPOSE AND INTRODUCTION TO THESIS

One of the major factors determining the effective operation of a gas turbine plant is the efficiency of the components-----turbine, compressor, and combustion chamber. Since the net work produced by such a unit is the difference between the turbine work and the compressor work, these component efficiencies must be kept as high as possible. Turbines and combustion chambers of relatively high efficiency can be designed; however, the design of highly efficient compressors is a major problem. The efficiency of a compressor is affected greatly by small changes in blade form. Since it is important that the efficiency of a compressor is not impaired during continuous operation, the effect of fouling on the blading becomes a major problem. In actual practice, a gas turbine plant under continuous operation may foul considerably due to the presence of foreign particles in the atmosphere. This fouling, in effect, will alter the shape of the compressor blades and thus subsequently reduce compressor efficiency. In the extreme case, all the turbine power output would go toward driving the compressor, leaving none for useful work. At sea, though the atmosphere is relatively free from dust particles, the presence of moist salt particles constitutes a source of fouling. It is essential, therefore, that the rate and magnitude of fouling be as accurately determined as possible, for, in the case of shipboard installations, the

operating life of a gas turbine unit, and thus the periods between overhauls, will depend largely thereupon.

The specific purpose of this thesis is to design and construct a suitable compressor test stand particularly adaptable to compressor fouling tests. Though conduction of these fouling tests will be the primary purpose of the unit, the design will lend itself readily to other compressor tests of a diversified nature.

Some work has been accomplished regarding the effect of wet compression in compressors, but this has been limited largely to centrifugal units. Water injection has been successfully employed in a few U.S.A.F. aviation gas turbine jets, but these were of the centrifugal type. A notable exception is the French Rateau SRA-101, which is equipped with a ten stage axial-flow compressor. Under takeoff conditions, with water injection into the compressor inlet, this unit develops 8820 pounds of thrust, an increase of 21% due to the water injection.

None of the research upon the effects of water injection has touched upon the effect of fouling on compressor performance. The presence of fouling on compressor blading has been noted from tests conducted at U. S. Naval Engineering Experimental Station, Annapolis, Maryland, but to this date steps to ascertain its resulting effects have not been undertaken.

During World War II and the years immediately preceding it, the Germans undertook an interesting series of

tests on axial-flow compressors. The results were published by Dr. Bruno Eckert in Stuttgart in 1946 at the request of the Naval Technical Mission in Europe. These were later translated by the Bureau of Aeronautics, Navy Department, and then published by the Bureau of Ships. The compressor performance results were much lower than those of current American and British designs, affected chiefly by excessive stage pressure rises and the ignoring of radial stability. However, the experimental techniques and theoretical analyses of the Germans were of unusual interest. The test rig consisted essentially of an open cycle compressor driven by an electric motor or a dynamometer, the whole unit being supported by a floating cradle. In one test rig, the air flow was controlled by a radial throttle at the compressor outlet, and metered by an orifice located ahead of the compressor inlet. Other test rigs placed the metering orifices at the compressor outlet and varied the air flow by using orifices of different diameters. Provisions were made for measuring the pressure and temperature at each stage. In addition, the compressor blades could be rotated to give any desired angle of attack.

Technical Note No. 1138 (National Advisory Committee for Aeronautics) entitled "Standard Procedures for Rating and Testing Multistage Axial-Flow Compressors" has been a very useful source of information for this thesis.

CHAPTER II - GENERAL DESCRIPTION OF TEST UNIT

The test stands described in the previous chapter were all of the open cycle design. For several reasons, however, the test unit finally decided upon for this thesis was a closed cycle design. For compressor fouling tests the closed cycle would provide better control of compressor inlet conditions------pressure, temperature and quantity of fouling material. It was decided to drive the compressor with the original turbine rather than with an electric motor, as this would eliminate shaft alignment difficulties and most of the bearing problems. The turbine would be driven by air from the wind tunnel, this, of course, resulting in a reduction from designed turbine power output. With the closed cycle the inlet conditions of the compressor would be kept at a partial vacuum, thus reducing the compressor work and increasing the maximum obtainable speed of the test unit.

The air flow through the test unit is divided into two distinct cycles----- the power air cycle and the compressor air cycle. See Figure I. The general arrangement of the unit is shown in Figure II. Fabrication of most of the ducting was accomplished at the Boston Naval Shipyard.

Power Air Cycle

The single stage turbine is driven by air from the supersonic wind tunnel. The air flows from the tunnel

outlet valve through a system of 12" ducting, a transition member narrowing to 8" piping, and thence into the duplex chamber. Sufficient flow area has been provided in the chamber to prevent the occurrence of high Mach Number air velocities. The air then expands through the turbine and exhausts through another transition member to the wind tunnel inlet valve. The power output of the turbine is varied by varying the air flow through the wind tunnel system.

Compressor Air Cycle

The compressor air cycle is designed as a closed cycle to operate at pressures somewhat below atmospheric. The air flows in a continuous cycle through the compressor inlet duct into the compressor, where it is compressed and exhausted into the duplex chamber. From there it is ducted through a system of piping to a gate valve. The latter can be adjusted to produce a wide range of air mass flow through the cycle. The air then passes through a transition flange to a 24" diameter elbow provided with air flow straighteners, whence it continues through a set of coolers followed by a wire screen and then reenters the compressor inlet duct, thus completing the cycle.

A means for exhausting the compressor air system has been provided. A line for this purpose has been installed in the transition member following the gate valve and leads to the exhauster system of the laboratory. Coupled with the exhauster system is a valve-controlled bleeder

system, bleeding air from the atmosphere to the compressor air cycle. The exhauster is operated at full capacity, and control of the air pressure at the compressor inlet duct is had by regulating the amount of air bled from the atmosphere through the bleeder system.

The temperature of the air entering the compressor is controlled by varying the water flow through the coolers.

PLATE A

General View of Test Facility

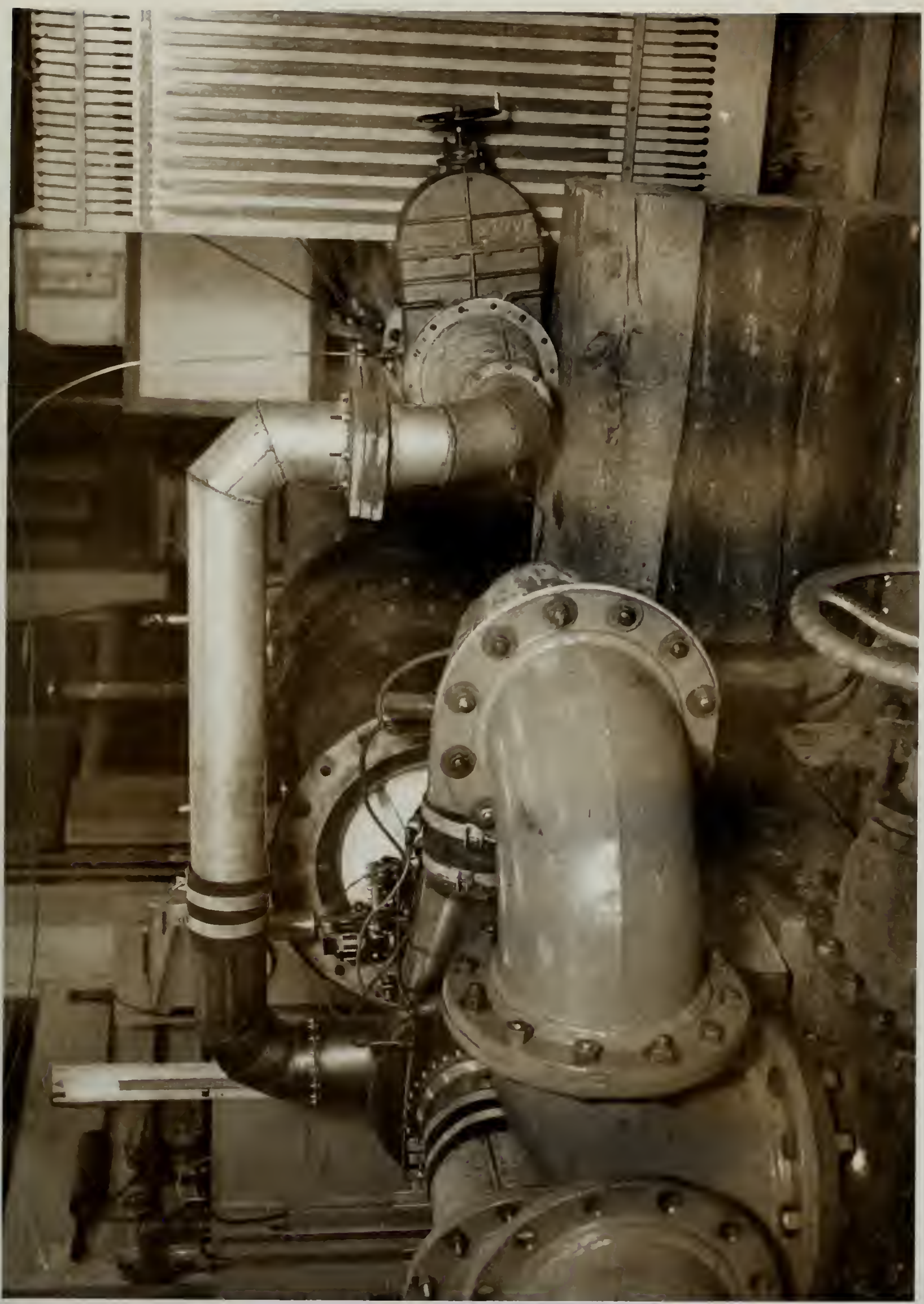


PLATE B

Test Facility - Right Side

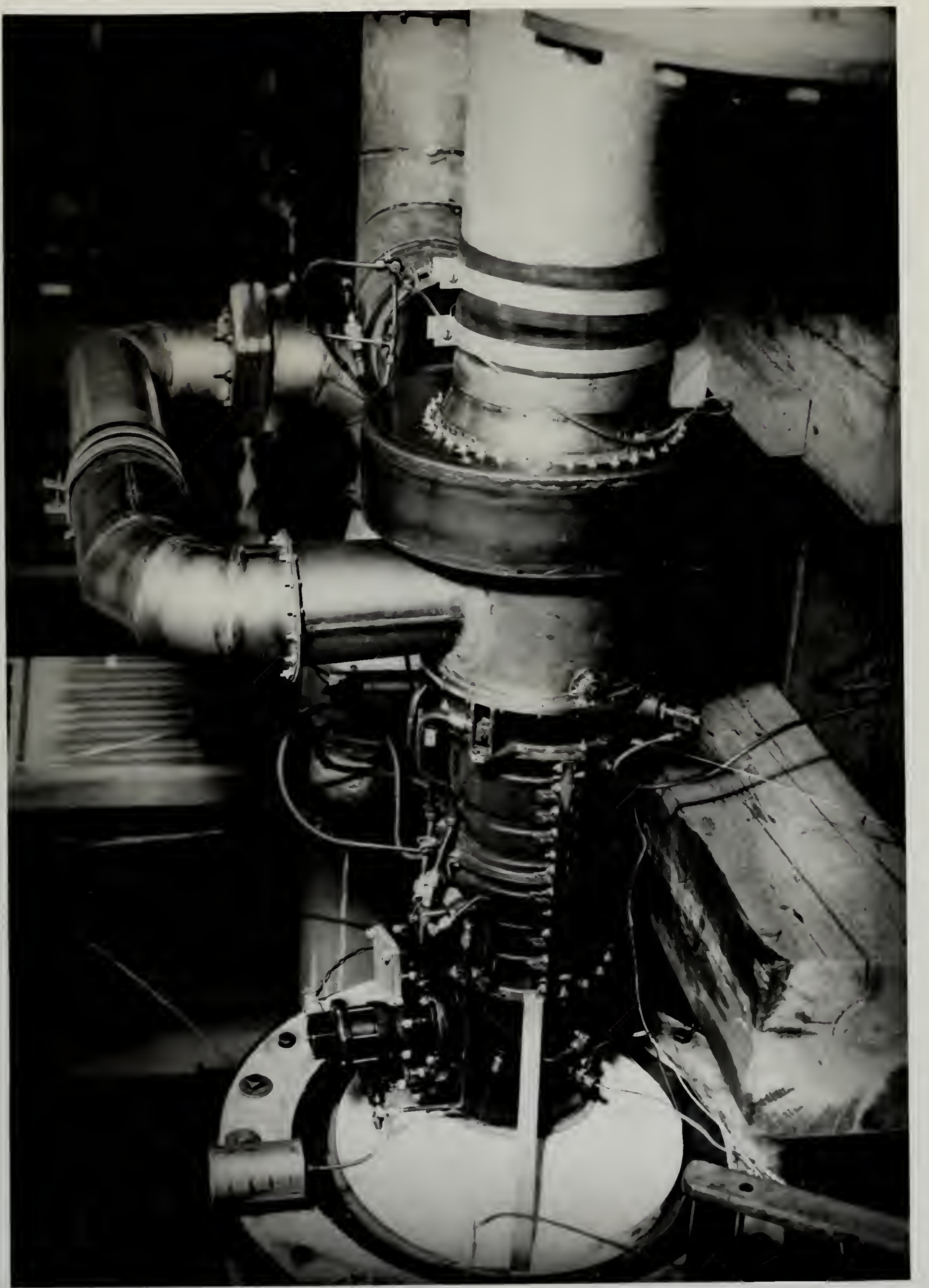


PLATE C

Test Facility - Left Side

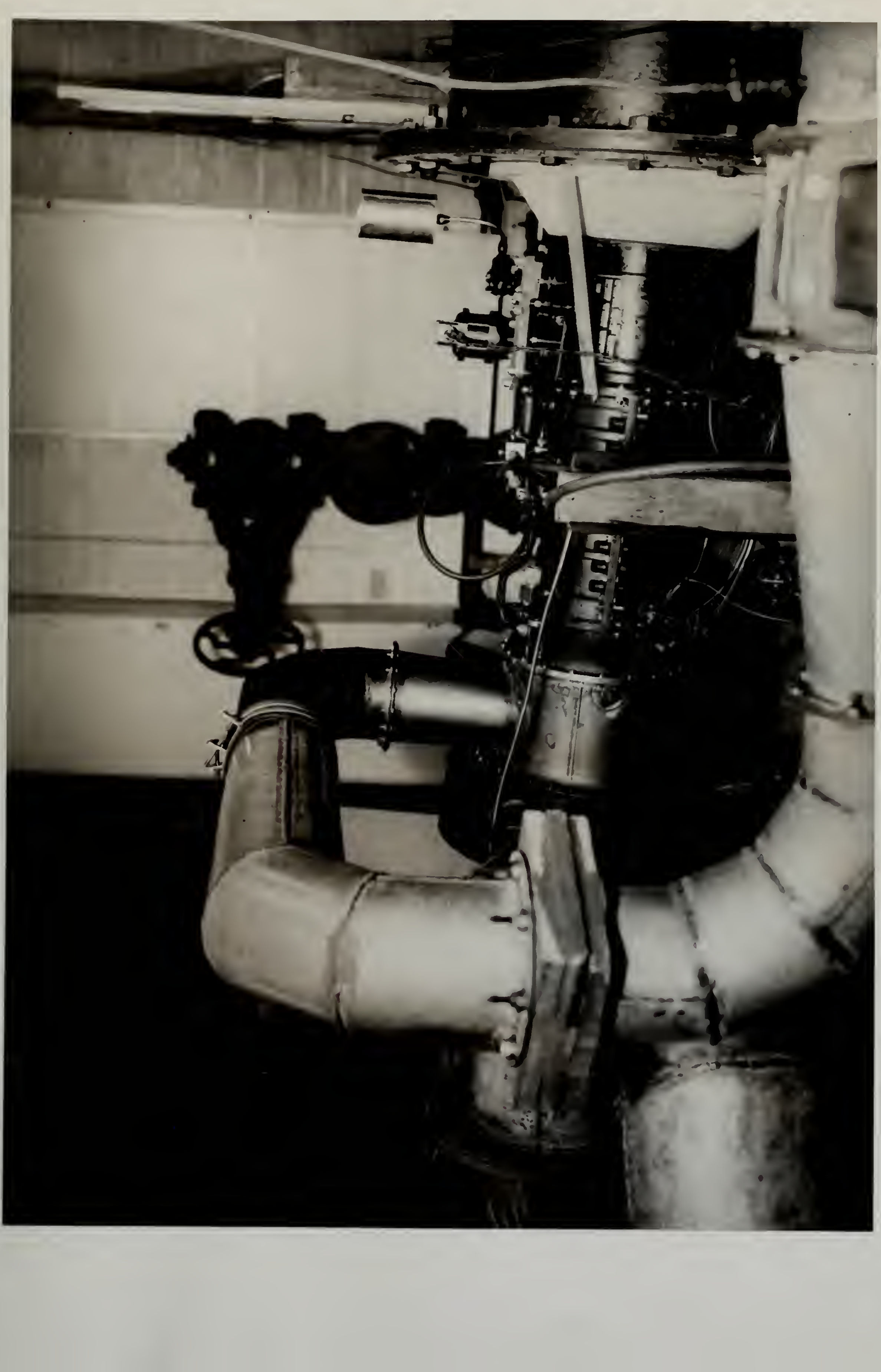
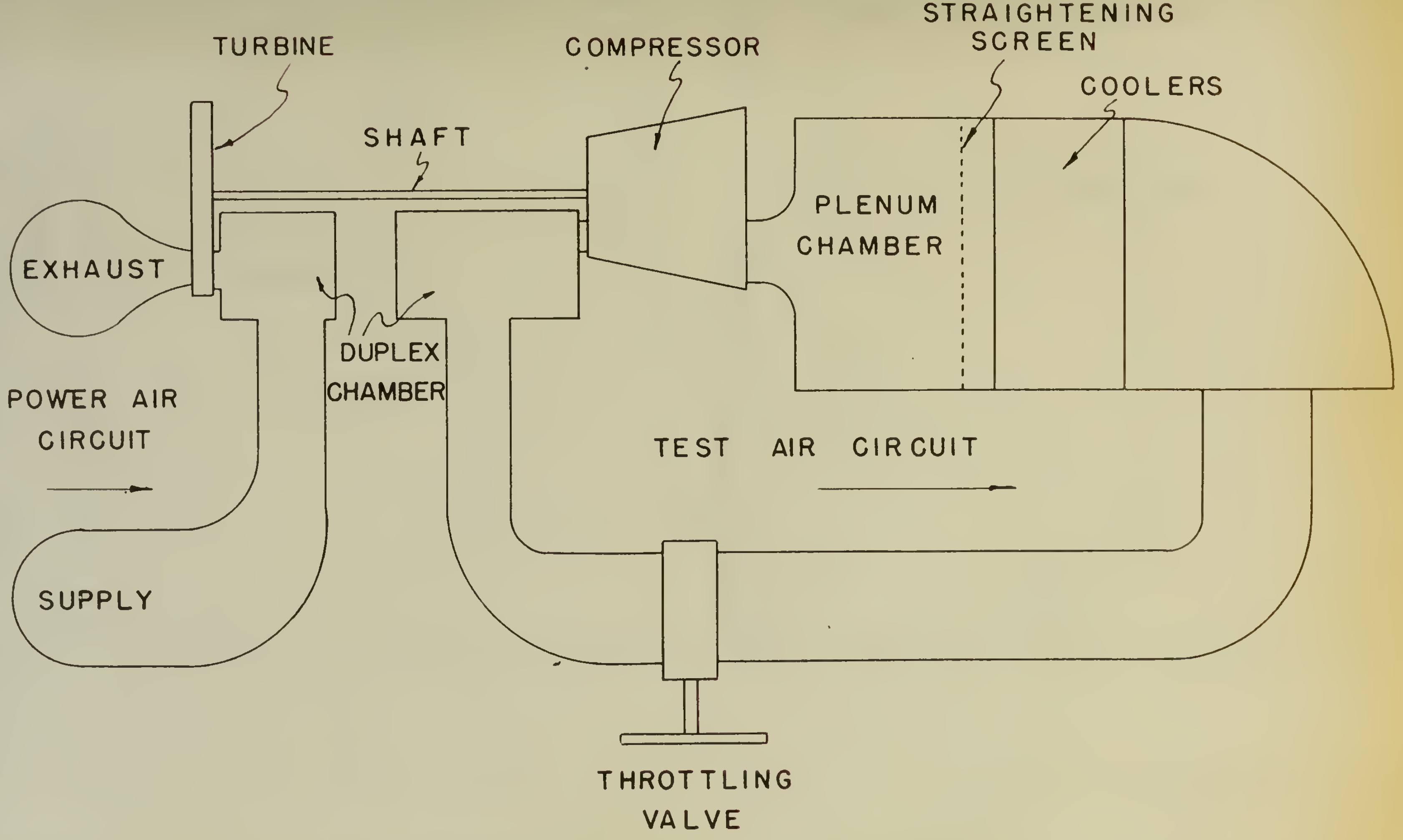
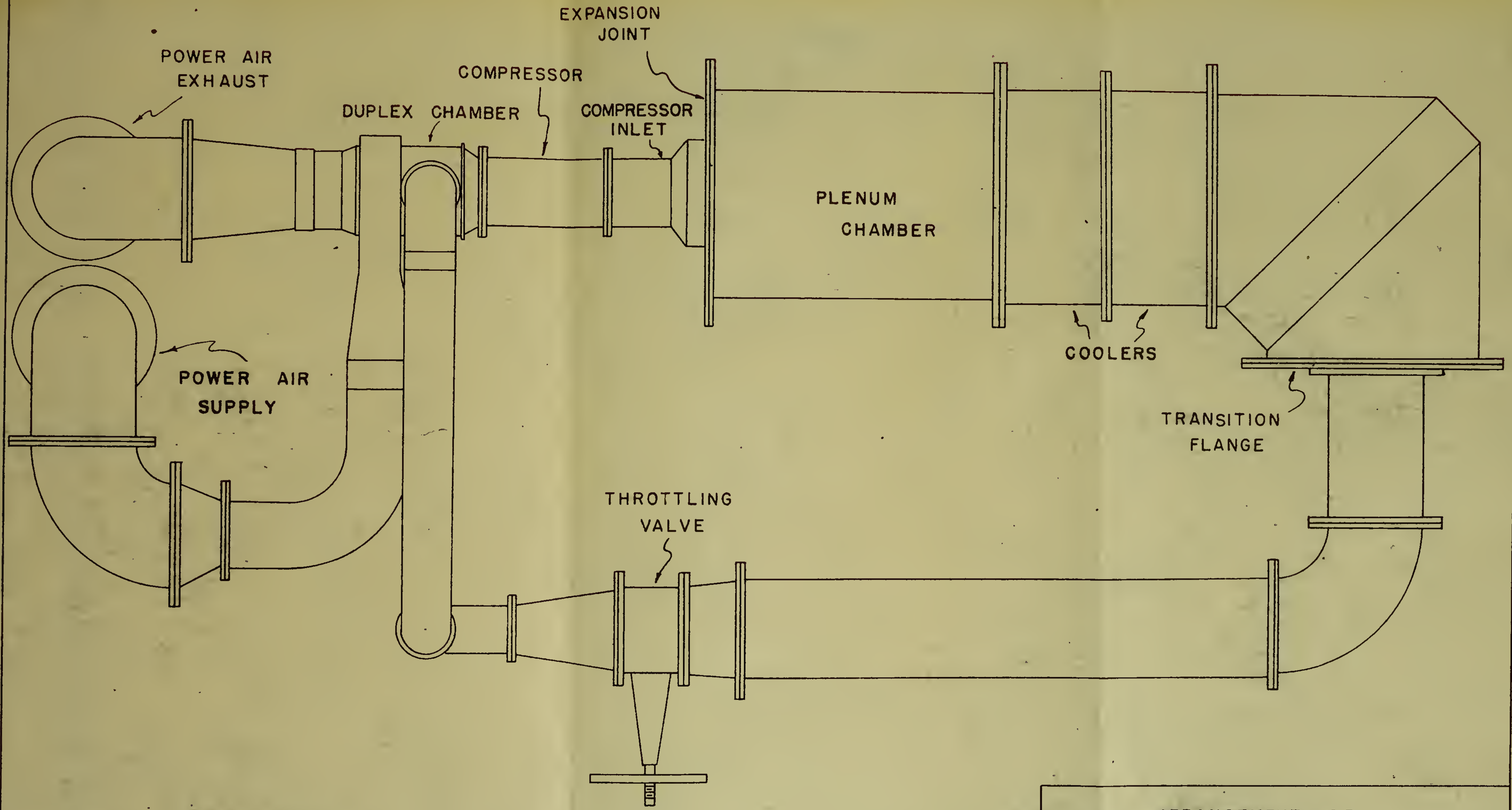


FIGURE I



SCHEMATIC PLAN VIEW
SHOWING
AIR CIRCUITS



ARRANGEMENT OF
EQUIPMENT FOR COMPRESSOR TEST
SCALE: 1" = 1'-0"

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CHAPTER III TESTING PROCEDURE FOR UNIT

The purpose of the test run was twofold. The main object, of course, was to test the functioning of the test stand, and correct any mechanical troubles that might become apparent. This was done by operating the compressor over a wide range of speed and pressure conditions. At the same time it was desired to obtain the operating characteristics of the compressor.

The speed of the compressor was controlled by regulating the speed of the wind tunnel compressor. The inlet and outlet wind tunnel valves were kept wide open at all times. The compressor speed was increased in increments of about 1000 rpm, with the following testing procedure used for each speed: With the compressor running at essentially constant speed, the air mass flow and compressor back pressure were varied by adjusting the gate valve in the compressor air cycle. Pressure, temperature and speed readings were taken with the gate valve fully open, about half open, one-quarter open, and until the compressor surge point was reached. When this procedure was completed, the speed of the compressor was increased, and the procedure cycle was repeated. The pressure in the plenum chamber was maintained as close to atmospheric pressure as

possible by controlling the quantity of air passing through the exhaustor and air bleeding system. Appendix D describes in detail the location and type of the instrumentation used for the test runs.

CHAPTER IV TEST RESULTS

Test data and results for the compressor test run of 9 May 1951, have been recorded in Tables I and II. Compressor characteristics as determined from the test data have been plotted in Figure IX, for the range of compressor speeds under 15000 RPM.

Due to the failure of the thermocouple wiring system, the compressor inlet temperature was approximated from the ambient air temperature and the air temperature leaving the compressor air cycle coolers. This was assumed for all calculations to be 70°F.

TABLE I

COMPRESSOR TEST DATA

DATE	9 MAY 1951
TEST NO.	1
CORR. BAR. PRESSURE	759.0 mm Hg
AMBIENT TEMPERATURE	83.3°F
WIND TUNNEL COOLER TEMP.	66° F

RUN	P_0 mm Hg	ΔP_{0-1} mm H_2O	P_{01} mm Hg	P_{02} mm Hg	$\Delta(P_{02}-P_2)$ mm H_2O	R P M
1	-24	122	-21	32	253	9650
2	-41	121	-22	31	118	9300
3	-41	116	-22	23	353	9700
4	-13	117	-21	28	353	10 100
5	-13	117	-22	29	356	10 100
6	-10	141	-22	51	503	12 000
7	-11	156	-21	49	485	12 000
8	-13	156	-21	52	442	11 800
9	-28	150	-21	60	325	11 200
10	-12	155	-21	51	476	12 000
11	-12	199	-21	66	645	13 650
12	-13	198	-19	68	657	13 700
13	-15	195	-21	72	613	13 450
14	-31	198	-20	76	524	13 200
15	-13	216	-21	72	696	14 300
16	-13	214	-21	68	627	14 500
17	-13	230	-20	76	711	14 850
18	-13	229	-21	78	691	14 950
19	-14	228	-21	79	685	14 850
20	-13	229	-21	84	695	14 850

COMPRESSOR CALCULATION SUMMARY

DATE 9 MAY 1951
TEST NO. 1
CORR. BAR. PRESSURE 759.0 mm Hg
AMBIENT TEMPERATURE 83.3°F
WIND TUNNEL COOLER TEMP. 66°F

RUN	CORR BAR PRESS mm Hga	ΔP INLET DUCT mm H_2O	P_0 mm Hga	T_1 °R	ΔP INLET DUCT mm Hg	P_1 mm Hga	P_{01} mm Hga	P_{02} mm Hga	$\Delta(P_{02}-P_2)$	P_2 mm Hga	P_2/P_1	N	$\frac{N}{\sqrt{T_1}}$	$\frac{100w\sqrt{T_1}}{P_1}$	
1	759	122	735	530	9	726	738	791	19	772	1.062	9650	419	7.02	
2	759	112	718	530	9	709	737	790	9	781	1.105	9300	403	6.70	
3	759	116	718	530	9	709	737	782	26	756	1.070	9700	420	6.88	
4	759	117	746	530	9	731	738	787	26	761	1.035	10100	439	6.90	
5	759	117	746	530	9	737	737	788	26	762	1.035	10100	439	6.90	
6	759	141	749	530	10	739	737	810	36	773	1.049	12000	520	7.55	
7	759	156	748	530	11	737	738	808	36	772	1.048	12000	520	7.93	
8	759	156	746	530	11	735	738	811	32	779	1.060	11800	512	7.90	
9	759	150	731	530	11	720	738	819	24	795	1.105	11200	486	7.77	
10	759	155	747	530	11	736	738	810	35	775	1.052	1200	521	7.90	
11	759	190	747	530	15	732	738	825	47	778	1.061	13650	592	8.81	
12	759	198	746	530	15	731	740	827	48	779	1.065	13700	595	8.81	
13	759	195	744	530	14	730	738	831	45	786	1.079	13450	584	8.80	
14	759	198	728	530	15	713	739	835	39	796	1.120	13200	573	8.81	
15	759	216	746	530	16	730	738	831	51	780	1.070	14300	620	9.24	
16	759	214	746	530	16	730	738	821	46	781	1.070	14500	630	9.18	
17	759	230	746	530	17	729	739	835	52	783	1.075	14850	645	9.53	
18	759	229	746	530	17	729	738	837	51	786	1.080	14950	650	9.50	
19	759	229	745	530	17	728	738	838	50	788	1.082	14850	645	9.48	
20	759	229	746	530	17	729	738	843	51	792	1.088	14850	645	9.50	

CHAPTER V DISCUSSION OF RESULTS

Being the first experimental run of the compressor test stand, the test results are more indicative than conclusive, and the accuracy of the data can be questioned The main object of the test run was achieved, since many interesting difficulties were brought to light.

The operation of the thermocouples was far from satisfactory. Readings of the potentiometer were erratic and at no time consistent. This was believed to be due to either a short circuit or a faulty connection in the thermocouple wiring system. For this reason, the compressor inlet temperature had to be inferred from the ambient air temperature and the air temperature at the outlet of the compressor air cycle cooler.

Compressor and turbine vibrations were considerable, particularly at two critical speeds - 250 RPM and 6000-7000 RPM. At 250 RPM, the bearing noise was considerable, being unusually loud and severe. Severe vibration of the unit at about 6000-7000 RPM was attributed to the struts supporting the compressor inlet duct. The identical trouble was recorded in the NACA logbook for this gas turbine unit. Extreme caution had to be exercised when recording total pressure readings, as the vibrations tended to alter the position of the pitot tube in the air flow. Because of the extreme

vibration, it was found almost impossible to keep the compressor at the surge point long enough to obtain pressure conditions and speed readings. It was feared that these vibrations might rupture one of the rubber diaphragms in the system.

The maximum speed reached on the test run was 15000 RPM, which is the rated idling speed of the gas turbine unit. Due to the uncertainty concerning the strength of the rubber diaphragms and also a shortage of testing time, it was decided to limit the test run to values below this speed.

As a result of this limitation in speed, the compressor characteristic curves plotted in Figure IX were limited to a very small portion of the total operating range of the compressor. The general trend of the curves were logical and indicated the approximate performance characteristics of the compressor, but the absolute values of the curves should not be considered as accurate.

The lubrication system seemed to function properly, and no serious trouble was experienced. The level of oil in the sump tank was maintained at a distance of about 3/8" from the bottom.

The test data for all runs below 9000 RPM were discarded when a faulty connection in the compressor inlet total pressure tap was discovered.

During the test run the compressor air cycle cooler developed a slight leak. A loose connection between the plenum chamber and the compressor inlet duct, probably caused by the vibration of the compressor inlet duct struts, might have had an effect on pressure readings.

The one compressor outlet temperature reading accurately measured was obtained with a mercury thermometer through an access plug in the duplex chamber.

CHAPTER VI CONCLUSIONS

From the results of the test runs the following conclusions may be drawn:

1. The design and construction of the compressor test stand is satisfactory, at present, for limited compressor tests. With the adaption of the recommendations stated in Chapter VII, a wider range of compressor tests will be possible.
2. The operating characteristics of the compressor were determined for a very limited range of speed and pressure conditions, below 15000 RPM. These results are merely indicative of the compressor characteristics, and should not be considered as accurate. Further tests should be conducted after design modifications have been completed.
3. The air mass flow through the compressor inlet duct can be measured accurately within 1.12%.

CHAPTER VII RECOMMENDATIONS

As a result of the experience gained during the test run, the following design modifications should be made before further testing of the unit:

(1) Replace the rubber diaphragms in the duplex chamber and between the compressor inlet duct and the plenum chamber with sheet metal diaphragms. With the present rubber diaphragms pressure variations at the compressor inlet are extremely limited and must be maintained at pressures near atmospheric.

(2) A thorough check of the turbine and compressor bearings should be made before operating at any speed higher than 15000 RPM.

(3) Replace the compressor inlet duct struts with struts of a heavier gauge sheet metal, or angle.

(4) A thorough check of the thermocouple circuit should be made. The thermocouples were not functioning properly, due probably to either a short circuit or a loose connection in the wiring system.

(5) Manometers of greater range should be used for monitoring of lubrication aspirating air and lubrication seal pressurizing line.

(6) Coolers should be checked for leaks - there apparently was a small leak in one of the coolers, although this may be of no consequence.

(7) A re-circulating line in the lubrication supply line should be installed, since the amount of lube oil pumped through the system to the aspirator depends on the temperature of the engine parts. This machine runs at relatively low temperatures and hence the lubrication needs are less.

(8) Due to small inaccuracies in the measurements and/or manufacture of the ducting, small misalignments should be corrected by lengthening or shortening ducting to fit.

(9) The operating characteristics of the compressor should be determined more accurately over the entire operating speed range. This was not possible previously due to factors discussed in Chapter V.

VIII. APPENDIX

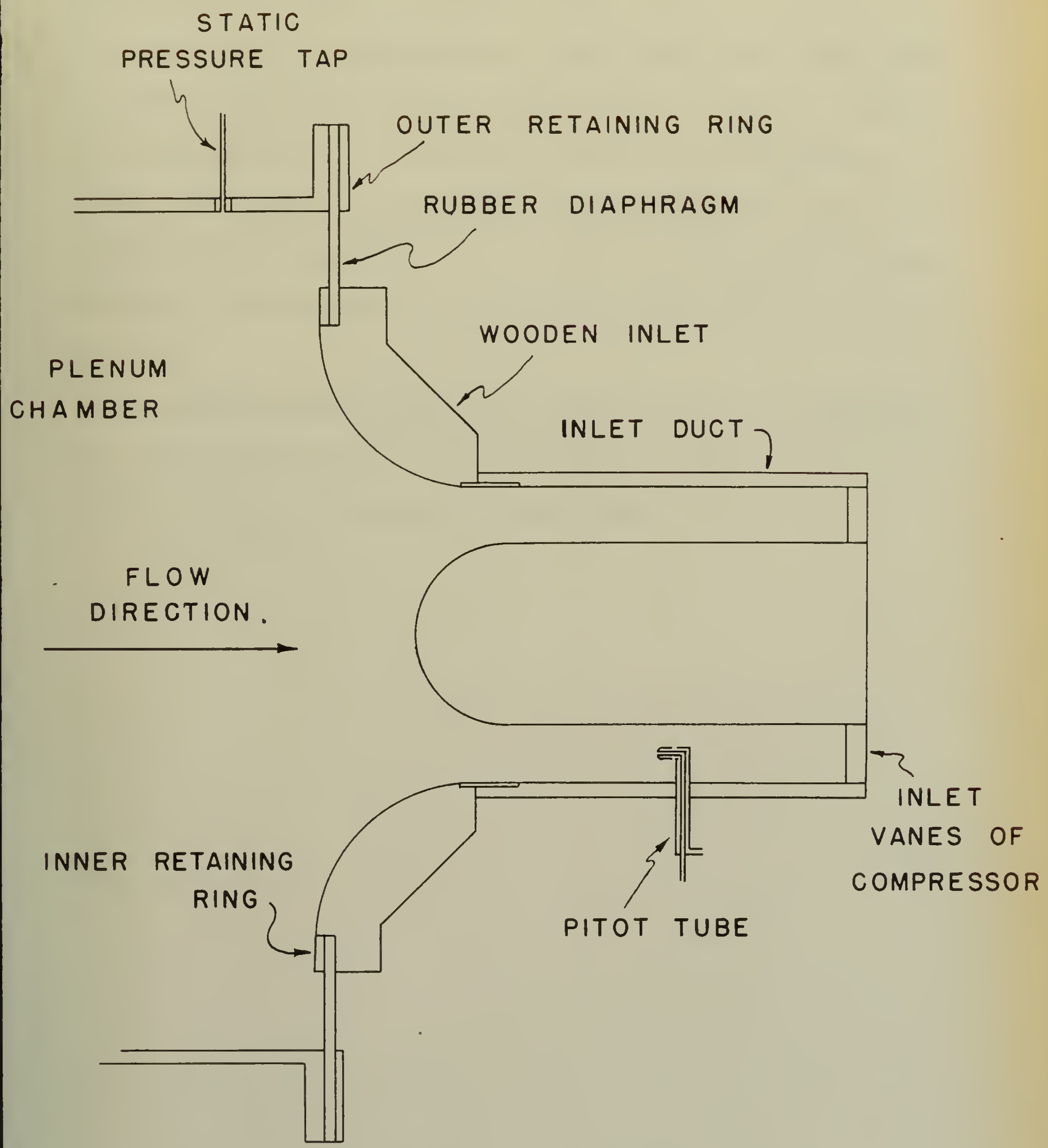
APPENDIX A - TURBINE AND COMPRESSOR ARRANGEMENT

The main component of the compressor test stand is the Westinghouse X9.5B aircraft turbojet which is designed for 200 pounds of thrust at military rating (36,000 RPM at static sea level conditions). The engine consists of an axial flow compressor of six stages with a maximum 3 to 1 pressure ratio at static sea level conditions, a double annular combustion chamber, and a single stage turbine. The unit has been modified to accommodate the power air cycle and the compressor air cycle by converting the combustion chamber into a duplex chamber. The two air cycles are separated in the duplex chamber by a two-ply 1/8" rubber diaphragm secured transversely across the chamber. A sectional view of the turbine and compressor units may be seen in Figure III. For general specifications of the unit reference is made to Westinghouse Electric Company Specification No. WAGT-X9.5-2 (Model Specification X9.5 Turbo-jet Engine).

The turbine, compressor, and duplex chamber are completely free of rigid connection to the remainder of the system-----accomplished by use of heavy rubber expansion joints. With this arrangement transmittal of ducting vibrations to the turbine and compressor will be kept to a minimum.

The compressor inlet duct, Figure IV, is secured to the plenum chamber by a 1/8" rubber diaphragm, the latter serving as an expansion joint. At its outer periphery, the

FIGURE IV



SCHEMATIC SIDE VIEW
COMPRESSOR
INLET SECTION

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diaphragm is secured to the plenum chamber flange with a steel retainer ring and bolted into position. At its inner periphery the diaphragm is made fast to the compressor inlet duct with a retainer ring and is held in position by wood screws and rubber cement. The compressor inlet duct, fabricated of white pine, is fitted with an aluminum ring designed to fit snugly within the compressor inlet; for rigidity, three sheet metal struts are employed.

To permit measuring of the mass flow to the compressor, the inlet duct has been calibrated. For detailed results reference is made to Appendix E.

APPENDIX B - DETAILED ARRANGEMENT OF DUPLEX CHAMBER

The design and construction of the duplex chamber was by far the most difficult phase of the thesis, limiting dimensions constituting the most serious problem. The basic design requirement stipulated that the chamber accomodate both the compressor and the power air flows. For a general arrangement of the chamber reference is made to Figures V and VI.

The air from the wind tunnel enters the turbine through the duplex chamber inlet annulus. The annulus, constructed of 12-gauge mild steel, of semi-weld construction, is designed to provide a uniformly distributed flow of air to the turbine nozzles. From the annulus the air enters the duplex chamber proper through six 2" by 3" ports located circumferentially around the outer shell of the chamber; these ports are of sufficient area to keep air velocities within acceptable limits-----below those corresponding to a Mach Number of 0.30 .

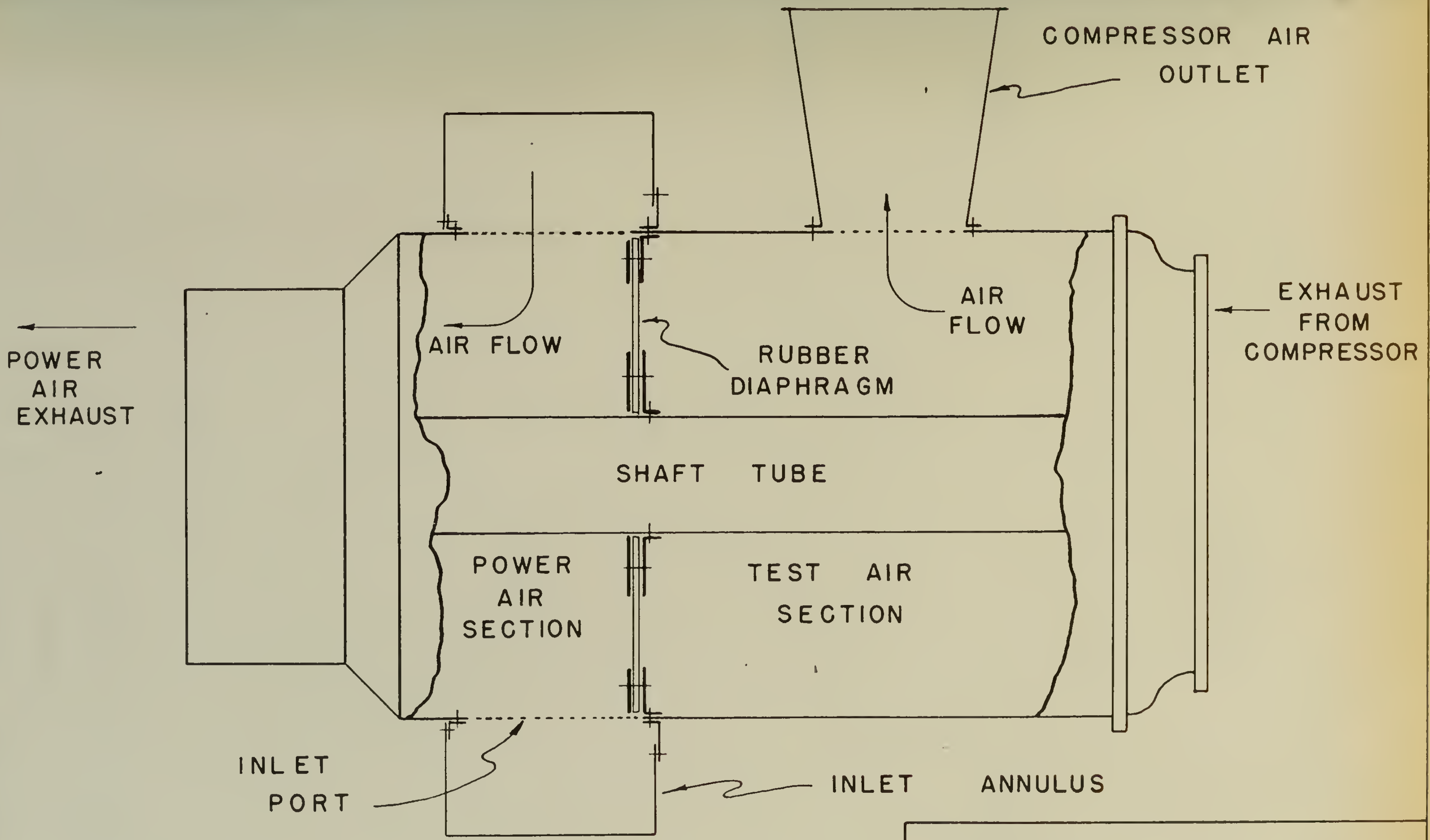
No welding operations were attempted on the shell of the chamber, this to prevent any misalignment of parts due to heat distortion. For this reason bolts are used for the securing of all parts. The inlet annulus is bolted to the outer shell of the chamber by means of two mild steel flanges. Rubber gaskets are employed to reduce air leaks to a minimum.

The compressor air outlet from the duplex chamber, of 16-gauge mild steel construction and 5 3/8" in diameter,

is bolted to the top of the chamber.

The rubber diaphragm separating the two air flows is held in position by two sets of retainer rings. One set is secured to the inside wall of the outer shell of the chamber, and the other set is secured to the outer wall of the inner shell. One ring in each set is flanged and bolted to the chamber shells. The diaphragm is then positioned and bolted in place between the two rings.

FIGURE V



SCHEMATIC SIDE VIEW
DUPLEX CHAMBER
INTERNAL ARRANGEMENT

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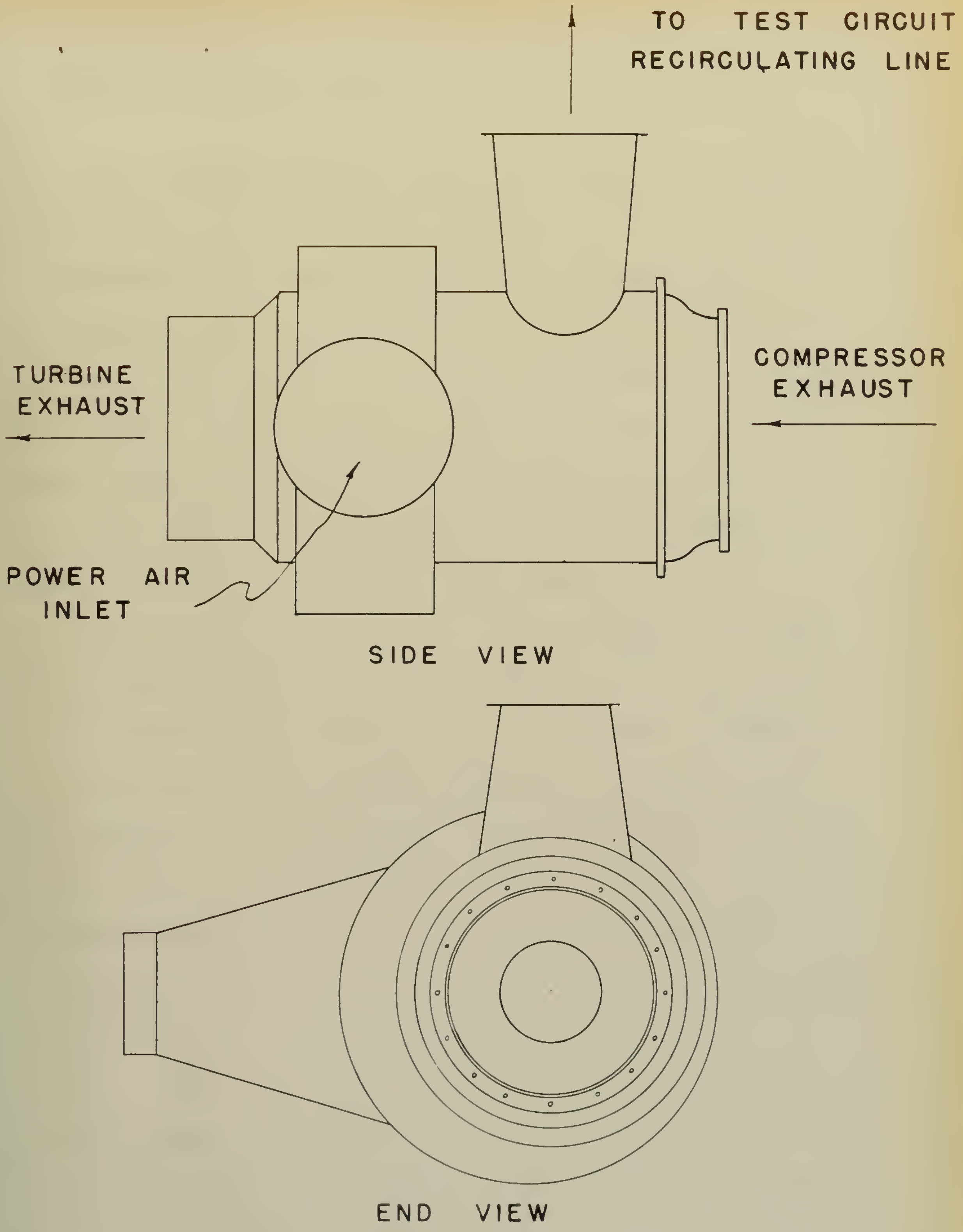
FIGURE VI



DUPLEX CHAMBER
SCHEMATIC VIEW
EXTERNAL ARRANGEMENT

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APPENDIX C - DETAILS OF THE LUBRICATION SYSTEM

Under normal operation, when the engine operates as a jet engine, the excess lubricant (oil mist) is allowed to enter the air stream and pass out of the engine with the exhaust gases of combustion.

When operating as a compressor test facility it will be important that no oil be permitted to enter the wind tunnel system to create serious fouling conditions. Accordingly, it was necessary to alter the lubrication system somewhat to meet this requirement.

Under normal operation (as a jet engine) , the lubricant exhausts into the air system in two locations:

(1) immediately forward of the compressor outlet

(2) between the turbine nozzles and turbine rotor

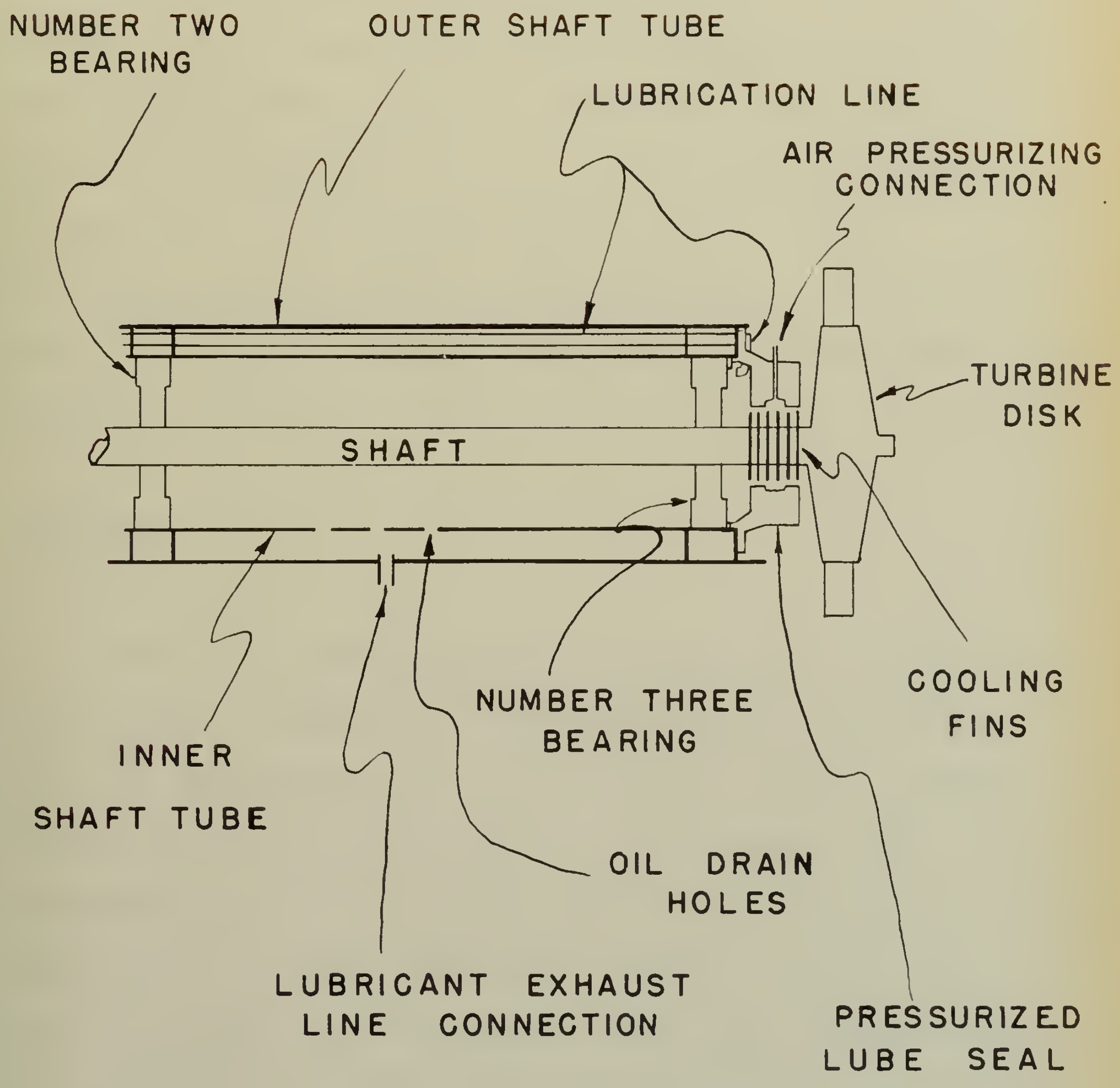
The prevention of oil leakage beyond the turbine has been accomplished in the following manner: A drain line has been installed leading from a point inside the inner shell of the duplex chamber, through a former fuel oil connection, and out to an oil trap and vacuum air pump. Thus, the drainage system has been so designed that a pressure differential exists between all parts of the unit and the drain line. To assist in the prevention of oil leakage between the turbine nozzles and blades (item 2, above) compressed air has been led to a bronze pressurized oil seal ring installed between the turbine bearing and turbine rotor as shown in Figure VII. The pressure gradient set up is, again, in the direction of the drain line. Thus, all oil

mist flow is in the direction of the drainage system.

To insure proper lubrication of the turbine bearing the lubrication line has been altered to run to a point aft of the bearing instead of before it as was the case originally. This permits oil flow in the direction of the pressure gradient mentioned above.

Installation of the bronze sealing ring made necessary the movement of the thermocouple for the turbine bearing to a point ahead of the bearing instead of after it.

FIGURE VII



SCHEMATIC SIDE VIEW
ALTERATIONS
LUBRICATION SYSTEM

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APPENDIX D INSTRUMENTATION OF THE TEST UNIT

The instrumentation is designed to measure: the pressure and temperature conditions at the inlet and outlet of the compressor; the inlet stagnation pressure and pressure drop across the compressor inlet duct, which has been calibrated as a fluid flow meter; the temperatures at the three turbine and compressor bearings; and the speed of the compressor. Figure X shows the general arrangement of the instrumentation system.

Pressure

The static pressure at the inlet of the compressor inlet duct is measured by a mercury vacuum manometer. The 1/8" pressure tap is located in the plenum chamber three inches from the compressor inlet duct. The pressure drop across the compressor inlet duct is measured on a water manometer. The other leg of the water manometer is connected to the static pressure side of a combination static-total pressure pitot tube located in the compressor inlet. The total pressure as measured by this pitot tube is indicated on a mercury manometer.

Another combination static-total pressure pitot tube is located in the compressor outlet. A mercury manometer measures the total pressure, while a water manometer is used to measure the pressure difference between the static and total pressure taps. All manometers are calibrated in millimeters.

Temperature

Iron-constantan 20 gauge wire thermocouples are used to determine temperature conditions in the system. A Leeds and Northrup double-range potentiometer indicator measuresthe thermocouple electromotive force. The thermocouple wires are connected at the temperature reference point, an ice bath contained in a thermos bottle, to copper lead wires from a multiple switch. A total of five thermocouples are used. Two thermocouples measure the stagnation temperatures at the compressor inlet and the compressor outlet. The other three thermocouples measure the lubricating oil temperatures at each of the three shaft bearings. The following color code was used for identification purposes - white or yellow indicating iron, and green or blue indicating constantan.

Speed

A tachometer connected to the main shaft through a bevel gear and operating at a speed ratio of 0.050 to 1 indicates the speed of the compressor rotor.

APPENDIX E CALIBRATION OF COMPRESSOR INLET DUCT

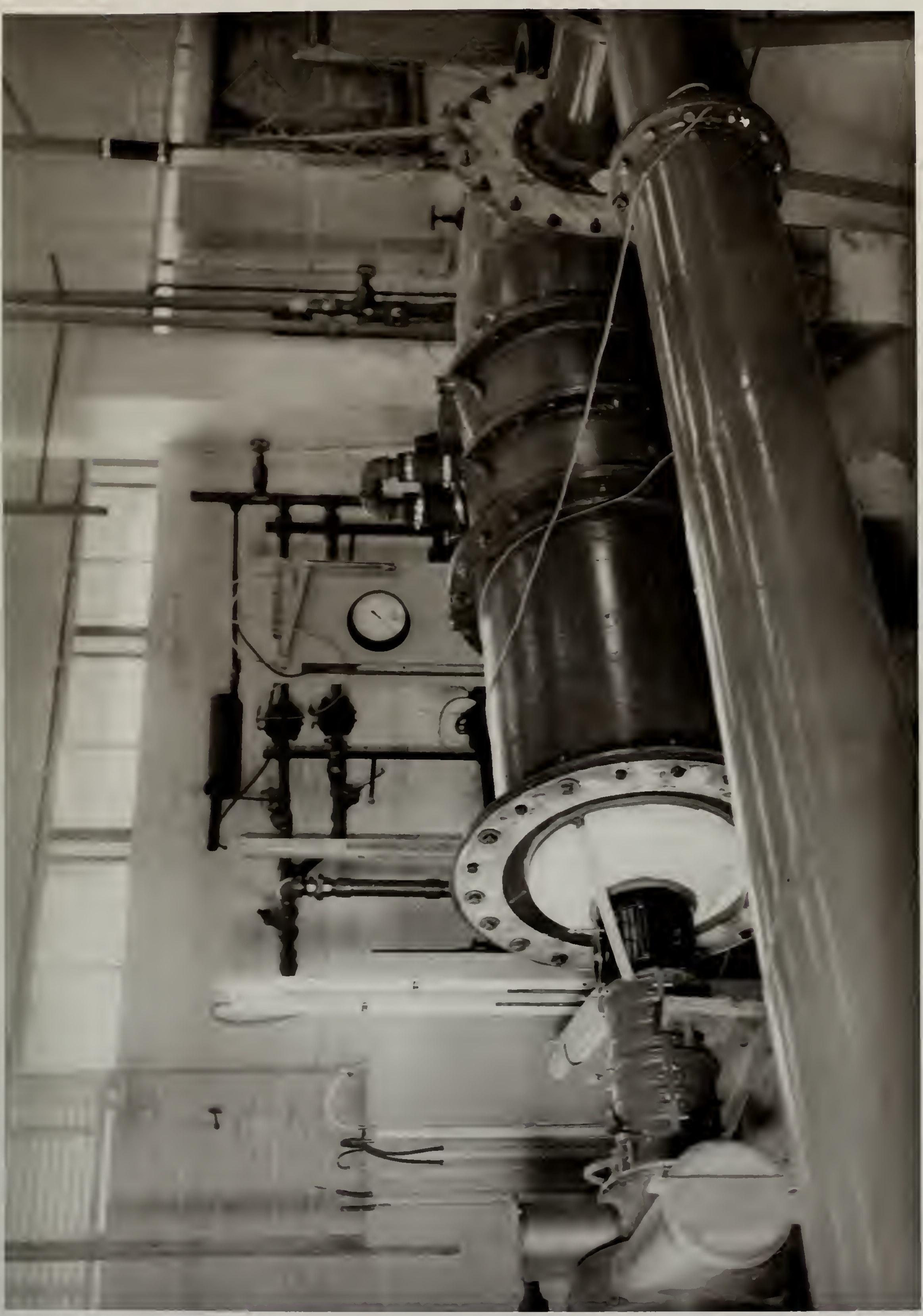
General Arrangement

To provide a means of determining the air mass flow through the compressor, the compressor inlet duct was calibrated as a flow meter with a standard ASME orifice plate. The arrangement of the test stand was modified for calibration purposes as shown in Figure VIII. A general view of the calibration layout is also shown in Plate D .

The principal alteration was that of combining the two closed air cycles to form a single flow of air from the wind tunnel through the system. For the calibration run the turbine and compressor rotors were removed, to eliminate any possible lubrication problems. The rubber diaphragm separating the two air cycles in the duplex chamber was removed, and all other outlets from the chamber were blanked off. The orifice used to calibrate the compressor inlet duct was a 8.25" standard ASME square edge orifice plate, with flange taps. To insure a smooth velocity profile of the air approaching the standard orifice, straightening tubes and a straightening screen were placed in the air ducting eight diameters in front of the orifice, as specified by ASME instructions. The standard orifice was positioned between two lengths of straight smooth ducting----8.6 diameters

PLATE D

General View Showing Calibration Set-up



in front of the orifice and 3.4 diameters behind the orifice. A wire screen straightener was placed in the plenum chamber to smooth out the air flow entering the compressor inlet duct. The water cooler air tubes in front of the screen also served to eliminate turbulence in the air flow.

Instrumentation

The primary measurements desired for the calibration were:

(1) The differential pressure across the orifice and across the compressor inlet duct.

(2) The static air pressure at the inlet of the standard orifice and at the inlet of the compressor inlet duct.

(3) The average air temperature across the standard orifice and the compressor inlet duct.

Pressure conditions on both sides of the standard orifice were measured from two flange taps, located one inch from each face of the orifice. A mercury manometer, calibrated in millimeters of mercury, measured the static pressure at the inlet of the orifice. The differential

pressure across the orifice was measured on a water manometer, calibrated in millimeters of water.

The static air pressure entering the compressor inlet duct was measured from an 1/8" static pressure tap, located in the plenum chamber, three inches from the inlet flange of the compressor inlet duct. A combination static-total pressure pitot tube located in the compressor intake was used in conjunction with the plenum chamber pressure tap to measure the differential pressure existing across the compressor inlet duct. The static air pressure was measured in millimeters of mercury; the differential pressure measured in millimeters of water. Calibration tests were run with two sizes of pitot tubes, with excellent correlation of results.

Temperature measurements were taken at the outlet of the main wind tunnel cooler. Since the temperature difference between the cooler temperature and the ambient room temperature was extremely slight, this temperature was sufficiently accurate for calibration purposes.

Calibration Procedure

Preparatory to the actual calibration run a 75 mm Hg. vacuum was impressed on the system to test for air leakage. After leakage had been eliminated the wind tunnel outlet valve was opened and the calibration

run was started. To prevent excessive pressure on the rubber diaphragm between the compressor inlet duct and the plenum chamber, the pressure in the plenum chamber was kept within 50 mm. Hg. of the atmospheric pressure. The air flow through the unit was changed by small intervals, and pressure readings taken after the flow had stabilized. This procedure was repeated over a wide range of air flows, limited only by the range of the standard orifice differential pressure manometer. A total of four runs was made. The calibration test data and test results appear in Tables III and IV. The method used in calculating these results is presented in Appendix F.

Plotting of Results

Instead of the usual plot of nozzle coefficient versus Reynolds Number, it was decided to plot the differential pressure across the compressor inlet duct versus the compressor flow parameter ---- $\frac{100\ w\ \sqrt{T_1}}{p_1}$

where T_1 is the compressor inlet temperature in °R.

w is the air flow in pounds per second

p_1 is the inlet duct inlet pressure in mm Hga.

This was deemed a more convenient method of plotting for use in determining the compressor characteristic curves. The calibration curve is shown in Figure XI.

Sources of Error

One of the assumptions made in the calculations of the air mass flow was that the flow through the compressor inlet duct was the same as that through the standard orifice, i.e. no air leakage occurred between the orifice and the compressor inlet duct. Thus any leakage would result in an error in determining air mass flow. This error would increase with increase in flow since air leakage increases with flow.

The mass flow calculations also assumed dry air flow. Any moisture in the air, resulting perhaps from wind tunnel cooler leakage, would affect the value of the calculated density. However, the density in the expression for the mass flow occurs as the square root, which should reduce this error to a negligible quantity.

The temperature of the air leaving the wind tunnel cooler was assumed to be the temperature at the orifice and the compressor inlet duct. This temperature should not be in error exceeding 3°R. or less than 1%.

Pressure readings were correct to the nearest millimeter of mercury or water, as the case may be. This introduces an average error not exceeding 0.13%.

Other errors include inaccuracy in reading charts, and slide rule errors. The total probable error is tabulated below.

	Limit of Error %	Square of Error %
Pressure --p	0.13	0.0169
" Δ p	0.13	0.0169
Temperature	0.58	0.3364
Density	0.05	0.0025
Primary element diameter	0.13	0.0169
Expansion factor--Y	0.50	0.2500
Thermal expansion	0.01	0.0001
Flow coefficient--K	0.50	0.2500
Sum of squares		0.8897
Tolerance (square root of sum of squares)		0.94%

Thus, the total error in calculating the mass flow through the standard orifice is less than 1%. Including similar errors in determining pressure and temperature conditions at the compressor inlet duct, the total error in calibration should not exceed - - 1.12%

Pressure --p	0.13	0.0169
" Δ p	0.13	0.0169
Temperature	0.58	0.3364
Total sum of squares		1.2599
Total error in calibration		1.12%

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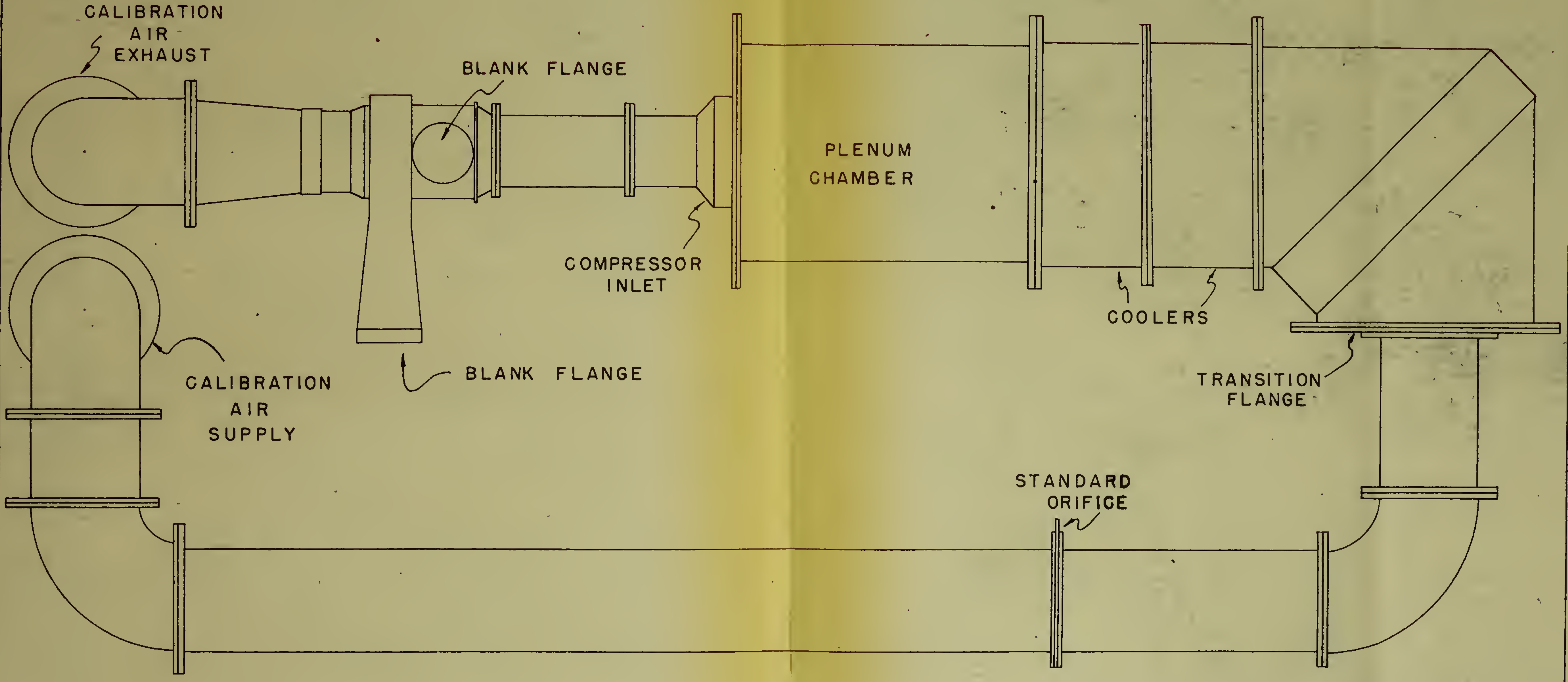

ARRANGEMENT OF
EQUIPMENT FOR CALIBRATION
SCALE: 1"=1'-0"

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TABLE III

CALIBRATION TEST DATA

Corrected Outside Barometric Pressure 30 546" Hga

Outside Ambient Temperature 75° F

Temperature Leaving Wind Tunnel Cooler 61° F

Type of Orifice ASME STANDARD 6 1/4" SQUARE EDGE ORIFICE WITH FLANGE TAPS

Test No. 1

Pitot Tube THICK

RUN NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
STANDARD ORIFICE INLET STATIC PRESSURE	-35	-29	-30	-30	-29	-28	-27	-27	-26	-25	-5	-2	0	+3	+5	+7
STANDARD ORIFICE ΔP	54	145	179	202	224	246	269	289	304	332	370	397	425	449	474	491
COMPRESSOR INLET INLET STATIC PRESSURE	-38	-38	-39	-41	-41	-42	-41	-41	-42	-42	-25	-21	-21	-20	-20	-18
COMPRESSOR INLET ΔP	29	77	96	97	102	111	122	130	140	150	166	179	192	202	215	224
RUN NO.	17	18	19	20	21	22	23	24	25	26	27	28	29			
STANDARD ORIFICE INLET STATIC PRESSURE	+7	+9	+11	+13	+16	+17	+17	+19	+21	+22	+22	+24	+24			
STANDARD ORIFICE ΔP	524	553	583	609	635	663	688	709	739	765	792	821	846			
COMPRESSOR INLET INLET STATIC PRESSURE	-19	-18	-18	-18	-18	-18	-18	-18	-18	-18	-18	-18	-18			
COMPRESSOR INLET ΔP	237	250	263	276	288	302	313	323	331	349	361	376	381			-

TABLE III (cont)

CALIBRATION TEST DATA

Corrected Outside Barometric Pressure 30.546" Hga

Outside Ambient Temperature 75° F

Temperature Leaving Wind Tunnel Cooler 61° F

Type of Orifice ASME STANDARD 6 1/4" SQUARE EDGE ORIFICE WITH FLANGE TAPS

Test No. 2

Pitot Tube THICK

RUN NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
STANDARD ORIFICE INLET STATIC PRESSURE	+16	+13	+11	+7	+5	+2	0	-1	-3	-6	-8	-10	-12	-18	-20	-22
STANDARD ORIFICE ΔP	788	769	673	621	563	512	462	413	362	314	272	228	182	97	42	20
COMPRESSOR INLET INLET STATIC PRESSURE	-25	-24	-25	-26	-26	-25	-25	-25	-25	-25	-24	-25	-24	-24	-24	-24
COMPRESSOR INLET ΔP	360	332	308	282	256	232	208	185	162	142	113	103	83	38	19	9
RUN NO.																

TABLE III (cont)

CALIBRATION TEST DATA

Corrected Outside Barometric Pressure 30.546" Hga

Outside Ambient Temperature 75° F

Temperature Leaving Wind Tunnel Cooler 61° F

Type of Orifice ASME STANDARD 6 1/4" SQUARE EDGE ORIFICE WITH FLANGE TAPS

Test No. 3 & 4

Pitot Tube THIN

RUN NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
STANDARD ORIFICE INLET STATIC PRESSURE	-26	-25	-24	-29	-22	-18	-10	-8	-6	-4	-1	+1	+5	+9	+12	+16
STANDARD ORIFICE ΔP	10	12	15	24	45	107	232	275	324	375	424	472	525	577	633	691
COMPRESSOR INLET INLET STATIC PRESSURE	-29	-27	-27	-27	-26	-25	-24	-24	-25	-25	-24	-24	-23	-24	-23	-21
COMPRESSOR INLET ΔP	6	7	8	11	19	48	102	122	153	180	207	233	264	291	319	348
RUN NO.	1	2	3	4	5	6	7	8	9	10	11					
STANDARD ORIFICE INLET STATIC PRESSURE	+17	+7	+1	-3	-12	-19	-21	-24	-28	-34	-31					
STANDARD ORIFICE ΔP	794	701	616	541	455	380	316	248	176	40	13					
COMPRESSOR INLET INLET STATIC PRESSURE	-23	-28	-30	-30	-36	-38	-38	-38	-38.0	-34	-32					
COMPRESSOR INLET ΔP	366	326	290	250	214	177	149	118	89	26	9					

TABLE IV

TEST NO. 1 RUN No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
CORR. BAR PRESS – mm Hg	775	775	775	775	775	775	775	775	775	775	775	775	775	775	775	775
ΔP ORIFICE mm H_2O	54	145	179	202	224	246	269	289	309	332	370	397	425	449	474	497
P_1 – ORIFICE – mm Hg a	740	746	745	745	746	747	748	748	749	750	770	773	775	778	780	782
AV. AIR TEMP – °R	521	521	521	521	521	521	521	521	521	521	521	521	521	521	521	521
ASSUMED K FOR $R_e = 5 \times 10^6$	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645
$\Delta P / P_1$ ORIFICE	.0054	0.014	0.018	0.020	0.022	0.024	0.027	0.028	0.030	0.033	0.035	0.038	0.040	0.043	0.045	0.047
Y	0.998	0.996	0.995	0.994	0.993	0.992	0.992	0.992	0.991	0.990	0.989	0.989	0.988	0.988	0.987	0.986
$\rho_1 = 0.05424 \, P_1 Y / T_1$	0.074	0.075	0.075	0.074	0.074	0.074	0.075	0.075	0.075	0.075	0.077	0.077	0.077	0.077	0.077	0.077
$w = 0.5048 \, Y \sqrt{\rho_1 \Delta P}$	1.001	1.650	1.830	1.940	2.040	2.140	2.235	2.320	2.400	2.475	2.645	2.745	2.845	2.920	3.004	3.080
μ_2	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}
$R_d = (2.04)\,10^5 \, w$	2.04×10^5	3.37×10^5	3.74×10^5	3.96×10^5	4.15×10^5	4.37×10^5	4.55×10^5	4.73×10^5	4.90×10^5	5.05×10^5	5.40×10^5	5.60×10^5	5.80×10^5	5.95×10^5	6.14×10^5	6.28×10^5
K	0.660	0.652	0.651	0.651	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650
W CORRECTED	1.025	1.670	1.850	1.960	2.060	2.160	2.255	2.340	2.420	2.495	2.670	2.775	2.870	2.945	3.035	3.105
P_1 – COMP. INLET – mm Hg	737	737	736	734	734	733	734	734	733	733	750	754	754	755	755	757
$\dfrac{100 w \sqrt{T_1}}{P_1}$	3.18	5.19	5.75	6.10	6.41	6.74	7.01	7.29	7.55	7.77	8.13	8.40	8.70	8.91	9.18	9.38
ΔP COMP INLET	29	77	76	97	102	111	122	130	140	150	166	179	192	202	215	224

TABLE IV (cont)

TEST NO. 1

Run No.	17	18	19	20	21	22	23	24	25	26	27	28	29			
Corr. Bar Press. - mm Hg	775	775	775	775	775	775	775	775	775	775	775	775	775			
ΔP orifice mm H_2O	524	553	583	609	653	665	688	709	734	765	792	821	846			
P_1 - orifice - mm Hg a	782	784	786	788	790	792	792	794	796	797	797	799	799			
Av Air Temp - °R	521	521	521	521	521	521	521	521	521	521	521	521	521			
Assumed K for $R_e = 5 \times 10^6$	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645			
$\Delta P / P_1$ orifice	0.049	0.052	0.055	0.057	0.059	0.062	0.064	0.066	0.068	0.071	0.073	0.0.76	0.078			
Y	0.984	0.984	0.983	0.983	0.982	0.982	0.981	0.980	0.980	0.979	0.978	0.977	0.977			
$\rho_1 = 0.05424\ P_1 Y / T_1$	0.077	0.077	0.078	0.078	0.078	0.078	0.078	0.078	0.078	0.078	0.078	0.078	0.078			
$w = 0.5048\, Y \sqrt{\rho_1 \Delta P}$	3.145	3.240	3.335	3.404	3.480	3.570	3.620	3.670	3.760	3.820	3.860	3.950	4.010			
μ_2	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}			
$R_d = (2.04)\, 10^5\, w$	6.90×10^5	6.60×10^5	6.80×10^5	6.95×10^5	7.10×10^5	7.29×10^5	7.39×10^5	7.46×10^5	7.67×10^5	7.79×10^5	7.87×10^5	8.05×10^5	8.15×10^5			
K	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650			
W corrected	3.175	3.265	3.360	3.440	3.510	3.600	3.650	3.700	3.745	3.855	3.890	3.980	4.050			
P_1 - comp. inlet - mm Hg	756	757	757	757	757	757	757	757	757	757	757	757	757			
$\dfrac{100\, w \sqrt{T_1}}{P_1}$	9.60	9.85	10.15	10.39	10.60	10.88	11.00	11.18	11.45	11.62	11.75	12.00	12.22			
ΔP comp inlet	237	250	263	276	288	302	313	323	337	349	361	376	387			

TABLE IV (cont)

TEST NO. 2 RUN No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
CORR BAR PRESS - mm Hg	775	775	775	775	775	775	775	775	775	775	775	775	775	775	775	775
ΔP ORIFICE mm H_2O	788	769	673	621	563	512	462	413	362	314	272	229	182	97	42	20
P_1 - ORIFICE - mm Hga	791	788	786	782	780	777	775	774	772	769	767	765	763	757	755	753
AV. AIR TEMP - °R	521	521	521	521	521	521	521	521	521	521	521	521	521	521	521	521.
ASSUMED h FOR $R_E = 5 \times 10^6$	0.645	0.645	0.645	0.645	0.645	0 645	0.645	0.645	0.645	0.645	0 645	0.645	0 645	0.645	0.645	0 645
$\Delta P / P_1$ ORIFICE	0.073	0.072	0.063	0.059	0.053	0.049	0.044	0.039	0.035	0 030	0.026	0 022	•018	0.009	0.004	0 002
Y	0 977	0.977	0.980	0.980	0.983	0.983	0.987	0.988	0.990	0.991	0.992	0.993	0 995	0.997	0.998	0 999
$\rho_1 = 0.05424 \, P_1 Y / T_1$	0.078	0.077	0.077	0.077	0.071	0.077	0.077	0.077	0.071	0.077	0.076	0.076	0.076	0.076	0.076	0.076
$w = 0.5048 \, Y \sqrt{\rho_1 \Delta P}$	3.855	3.800	3 565	3.420	3.260	3.100	2.970	2.800	2.625	2.445	2.283	2.090	1.810	1.375	0.895	0.619
μ_2	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}
$R_d = (2.04)\, 10^5 \, w$	7.86×10^5	775×10^5	7.27×10^5	$6\,96 \times 10^5$	$6\,65 \times 10^5$	6.31×10^5	6.06×10^5	5.71×10^5	$5\,36 \times 10^5$	4.99×10^5	4.65×10^5	4.26×10^5	3.81×10^5	2.80×10^5	1.83×10^5	1.26×10^5
K	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.651	0.653	0.655	0.657
W CORRECTED	3.880	3.835	3.600	3 445	3.285	3.125	2.995	2 820	2.645	2.465	2.315	2.105	1.890	1.390	0.909	0.630
P_1 - COMP. INLET - mm Hg	750	751	750	749	749	750	750	750	750	750	751	750	751	751	751	751
$\dfrac{100 \, w \sqrt{T_1}}{P_1}$	11.81	11.65	10.98	10.52	10.03	9.52	9.11	8.58	8.06	7.51	7.34	6.41	5 75	4.22	3.02	1.92
ΔP COMP INLET	360	332	308	282	256	232	208	185	162	142	113	103	83	38	19	9

TABLE IV (cont)

TEST NO. 3

RUN NO	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
CORR BAR PRESS mm Hg	775	775	775	775	775	775	775	775	775	775	775	775	775	775	775	775
ΔP ORIFICE mm H_2O	10	12	15	24	45	107	232	275	324	375	424	472	525	577	633	691
P_1 - ORIFICE - mm Hga	749	750	751	751	753	757	765	767	769	771	774	776	780	784	787	791
AV AIR TEMP °R	521	521	521	521	521	521	521	521	521	521	521	521	521	521	521	521
ASSUMED K FOR $R_E = 5 \times 10^6$	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645
$\Delta P / P_1$ ORIFICE	0.001	0.001	0.001	0.002	0.004	0.010	0.023	0.026	0.031	0.036	0.040	0.045	0.050	0.054	0.059	0.064
Y	1.000	1.000	1.000	1.000	0.999	0.997	0.993	0.991	0.991	0.989	0.988	0.985	0.984	0.982	0.981	0.979
$\rho_1 = 0.05424 \, P_1 Y / T_1$	0.075	0.075	0.076	0.075	0.076	0.076	0.076	0.077	0.076	0.077	0.078	0.077	0.077	0.077	0.078	0.078
$w = 0.5048 \, Y \sqrt{\rho_1 \Delta P}$	0.436	0.480	0.536	0.677	0.967	1.430	2.105	2.285	2.495	2.680	2.860	2.995	3.155	3.300	3.455	3.605
μ_2	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}
$R_d = (2.04) \, 10^5 \, w$	0.89×10^{5}	0.98×10^{5}	1.09×10^{5}	1.38×10^{5}	1.89×10^{5}	2.92×10^{5}	4.3×10^{5}	4.67×10^{5}	5.09×10^{5}	5.46×10^{6}	5.82×10^{5}	6.10×10^{5}	6.44×10^{5}	6.74×10^{5}	7.05×10^{5}	7.36×10^{5}
K	0.665	0.662	0.661	0.657	0.655	0.652	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650
w CORRECTED	0.450	0.495	0.550	0.690	0.941	1.445	2.123	2.280	2.515	2.700	2.885	3.020	3.180	3.330	3.480	3.640
P_1 - COMP INLET mm Hg	746	748	748	748	749	750	751	751	750	750	751	751	752	751	752	754
$\frac{100 \, w \sqrt{T_1}}{P_1}$	1.38	1.51	1.68	2.11	2.88	4.40	6.47	6.94	7.65	8.22	8.79	9.20	9.66	10.12	10.59	11.05
ΔP COMP INLET	6	7	8	11	19	48	102	122	153	180	207	233	264	291	319	348

TABLE IV (cont)

TEST NO. 4 RUN No	1	2	3	4	5	6	7	8	9	10	11					
CORR BAR PRESS mmHg	775	775	775	775	775	775	775	775	775	775	775					
ΔP ORIFICE mmH$_2$O	794	701	616	541	455	380	316	248	176	40	13					
P_1 - ORIFICE - mm Hga	792	782	776	772	763	756	754	751	747	741	744					
AV AIR TEMP °R	521	521	521	521	521	521	521	521	521	521	521					
ASSUMED K FOR $R_E = 5 \times 10^5$	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645	0.645					
$\Delta P / P_1$ ORIFICE	0.074	0.066	0.059	0.052	0.044	0.037	0.031	0.024	0.011	0.004	0.001					
Y	0.976	0.979	0.981	0.984	0.989	0.990	0.992	0.995	0.995	0.999	1.000					
$\rho_1 = 0.05424 \ P_1 Y / T_1$	0.078	0.077	0.077	0.076	0.076	0.075	0.075	0.075	0.075	0.075	0.074					
$w = 0.5048 \ Y \sqrt{\rho_1 \Delta P}$	3.860	3.605	3.400	3.180	2.915	2.655	2.430	2.160	1.820	0.869	0.495					
μ_2	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}	1.2×10^{-5}					
$R_d = (2.04)\ 10^5\ w$	7.88×10^5	7.35×10^5	6.93×10^5	6.5×10^5	5.93×10^5	5.41×10^5	4.95×10^5	4.41×10^5	3.70×10^5	1.78×10^5	1.01×10^5					
K	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.650	0.651	0.655	0.667					
w CORRECTED	3.890	3.635	3.420	3.205	2.935	2.675	2.450	2.175	1.835	0.881	0.513					
P_1 - COMP INLET mmHg	752	747	745	745	739	737	737	737	737	741	743					
$\dfrac{100\ w \sqrt{T_1}}{P_1}$	11.82	11.10	10.49	9.82	9.06	8.30	7.60	6.75	5.70	2.72	1.58					
ΔP COMP INLET	366	326	290	250	214	177	149	118	89	26	9					

APPENDIX F SAMPLE CALCULATIONS

Calibration

D_1 - actual inside diameter of ducting - - - - - 10.30"

D_2 - primary element, standard orifice - - - - - 6.25"

Diameter ratio, D_2/D_1 = (6.25)/(10.30) - - - - - 0.595

Corrected barometric pressure = 30.55"Hga - - -775.0 mm Hga

Differential pressure across orifice, as

measured from flange taps, Δp - - - - - - - - 54.0 mm H_2O

Static pressure at orifice inlet as measured - (-35.0)mm Hg

Average air temperature at orifice, T_1 - - - - - 521°R

Assume flow coefficient, K, from Figure 34* - - 0.645

Static pressure at orifice inlet, p_1 =

(775.0 mm Hga - 35 mm Hg) - - - - - - - - - - -740.0 mm Hga

$\Delta p/p_1$ = (54)(0.0737)/(740) - - - - - - - - - - - - 0.00539

Expansion factor, Y, from Figure 37* - - - - - - - 0.998

Density, assuming dry air:

$\rho_1 = 2.70 (p_1Y/T_1)$ where p_1 = psi

$\rho_1 = 0.0524(p_1Y/T_1)$ where p_1 = mm Hga

$\rho_1 = 0.0524(740)(0.998)/(521)$ - - - - - - 0.074 lbs/ft^3

Air mass flow, w, in pounds per second:

$w = 0.668\ A_2\ K\ E\ Y\ \sqrt{\rho_1\ \Delta p}$

where A_2 - throat area of orifice in sq. in.

K - assumed to be 0.645

E - area multiplier for thermal

expansion, from Figure 3* - 1.0

Y - empirical expansion factor

ρ_1 - pounds per cubic foot

Δp - pounds per square inch

$$w = (0.668)(30.68)(1.0)(0.645)\ Y \sqrt{\rho_1 \Delta p (0.001421)}$$

$$w = (0.5048)\ Y \sqrt{\rho_1 \Delta p} \qquad (\text{where } \Delta p = \text{mm } H_2O)$$

First trial for air mass flow:

$w = (0.5048)(0.993)(0.074)(54) =$ - - - - - - - - 1.001 lbs/sec

Absolute viscosity, μ_2, from Figure 14* - - - - - - - $(1.2) \times 10^{-5}$

$$\text{Reynolds Number} = \frac{48w}{\pi D_2 \mu_2}$$

$$= \frac{48w \cdot 10^{+5}}{(3.14)(6.25)(1.2)}$$

$$= (2.04) \times 10^5 \times w$$

$= (2.04)(10^5)(1.001)$ - - - - - - - 2.04×10^5

Actual K value from Figure 34* - - - - - - - - - - - - 0.654

Corrected air flow = $\frac{(0.654)}{(0.645)} \times (1.001)$ - - - - - 1.015 lbs/sec

Static pressure at compressor inlet duct, p_c

$=$ (-38) mm Hg - - - - - - - - - - - - 737.0 mm Hga

Weight flow parameter, $\frac{100 w \sqrt{T_c}}{p_c}$, at compressor

inlet duct - 3.14

Differential pressure across compressor inlet duct- 29 mm H_2O

---

* - Figures referred to are found in Flow Measurement, 1949.

Compressor Characteristics

Due to the failure of the thermocouple circuit during the test runs, the temperature at the inlet of the compressor was inferred from the ambient air temperature and the temperature at the compressor air cycle cooler. It was assumed to be 70 °F.

From test run #15, the following data was obtained:

Corrected barometric pressure = 28.85"Hga = 759.0 mm Hga

Inlet duct inlet pressure - - - - - - - - - - -13.0 mm Hg

Differential pressure across compressor
inlet duct, Δp_{o-1} - - - - - - - - - - 216.0 mm H_2O

Compressor inlet total pressure - - - - - - -21.0 mm Hg

Compressor outlet total pressure - - - - - 72.0 mm Hg

Differential pressure, static-total compressor outlet pressures = $\Delta(p_{O2} - p_2)$- 696.0 mm H_2O

Speed - 14,300 RPM

From the above data:

Inlet duct inlet pressure = 759.0 - 13.0 = p_0 746.0 mm Hga

Compressor inlet static pressure = p_1 =
746.0 - 216(0.07349) = - - - - - - - - - 730.0 mm Hga

Compressor outlet total pressure = p_{O2} =
759.0 + 72.0 = - - - - - - - - - - - - - - 831.0 mm Hga

---

* - 1 mm H_2O equals 0.07349 mm Hg

Compressor outlet static pressure = p_2 =

831.0 - 696(0.07349) = - - - - - - - - - - 780.0 mm Hga

Compressor pressure ratio = p_2/p_1 - - - - - - 1.07

From compressor inlet duct calibration curve,

for Δp = 216.0 mm H_2O $\dfrac{w\sqrt{T_1}}{p_1}$ = - - 9.24

$\dfrac{N}{\sqrt{T_1}} = \dfrac{14{,}300}{\sqrt{530}}$ = - - - - - - - - - - - - - 620.0

APPENDIX G BIBLIOGRAPHY

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FIGURE IX

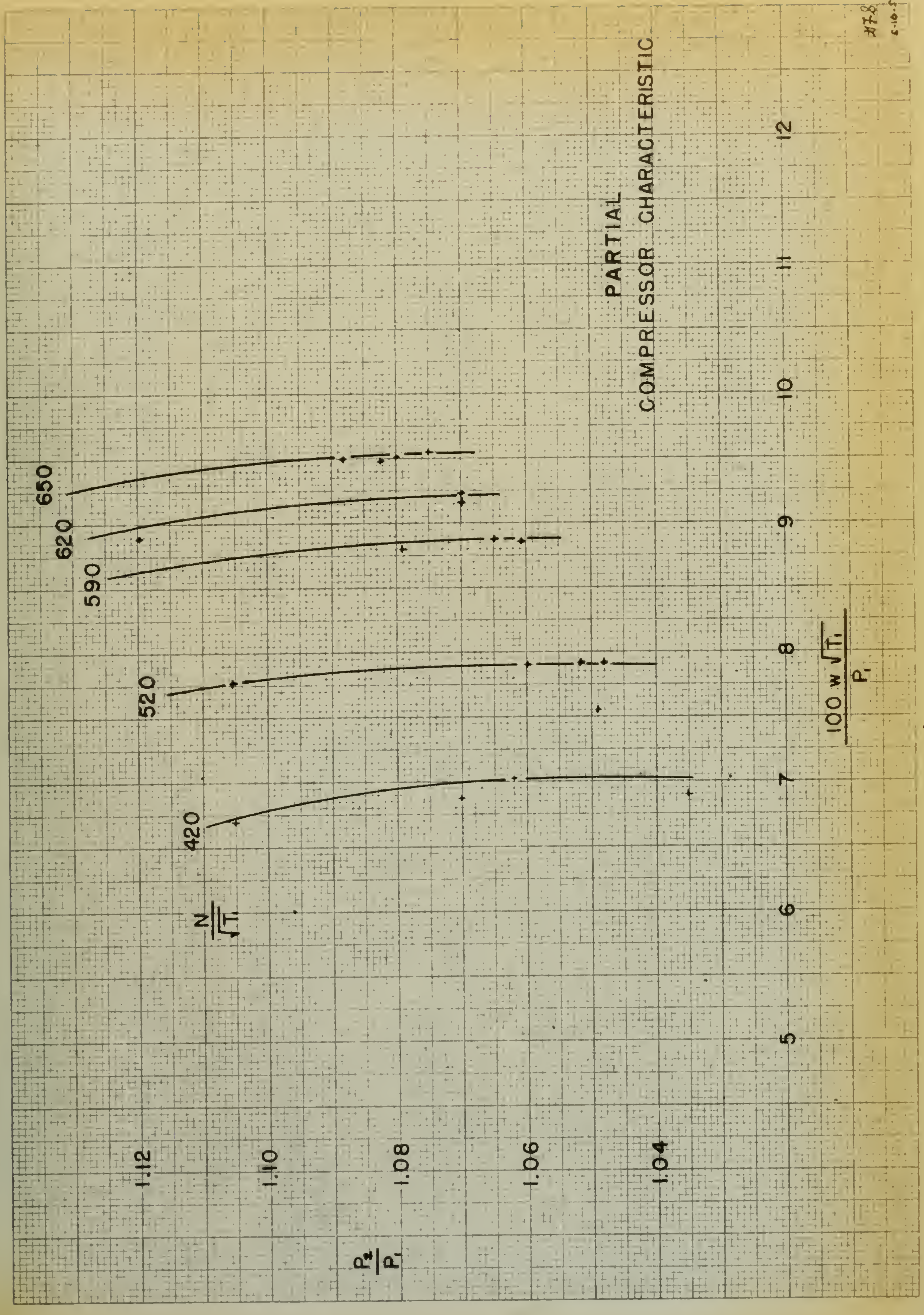
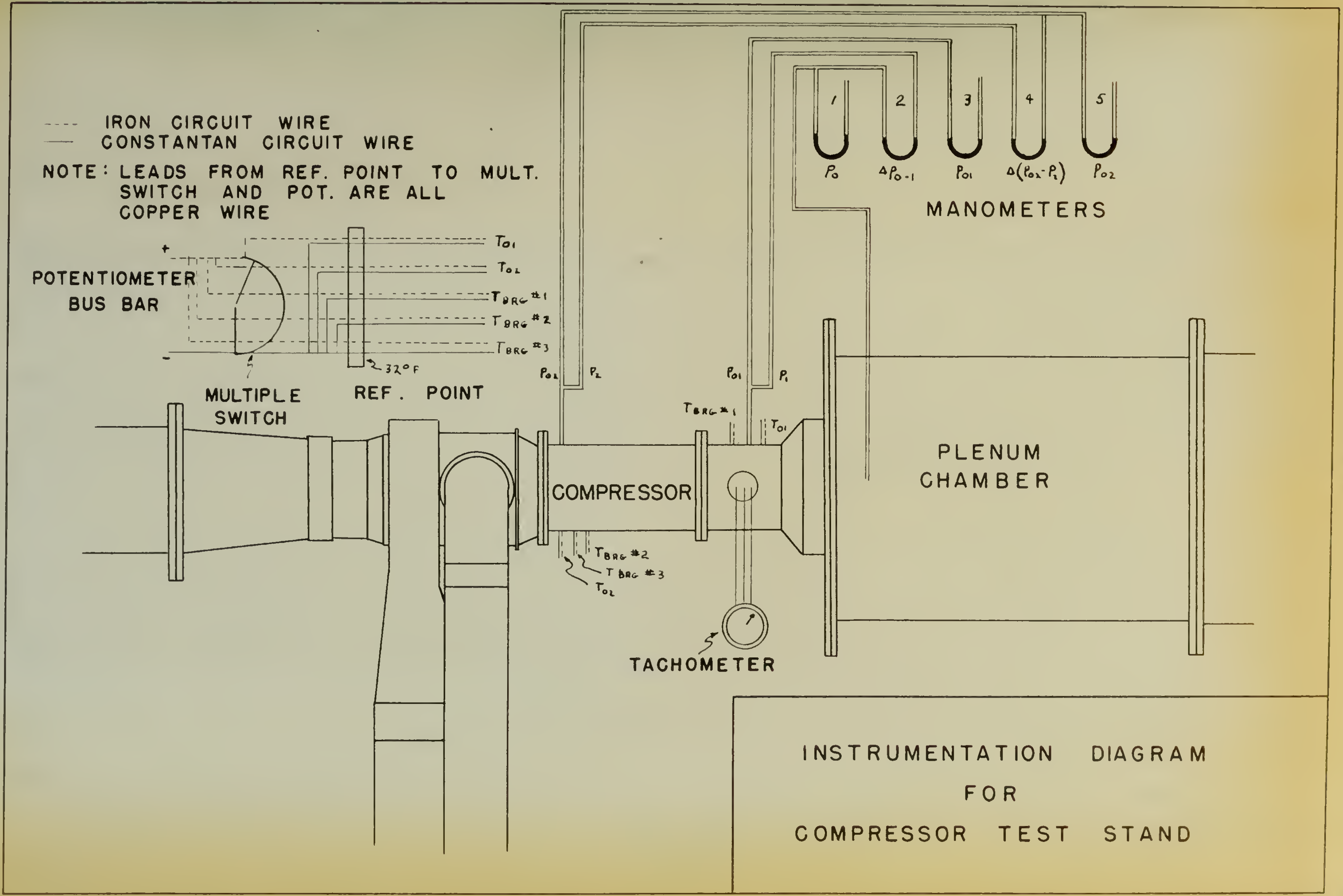
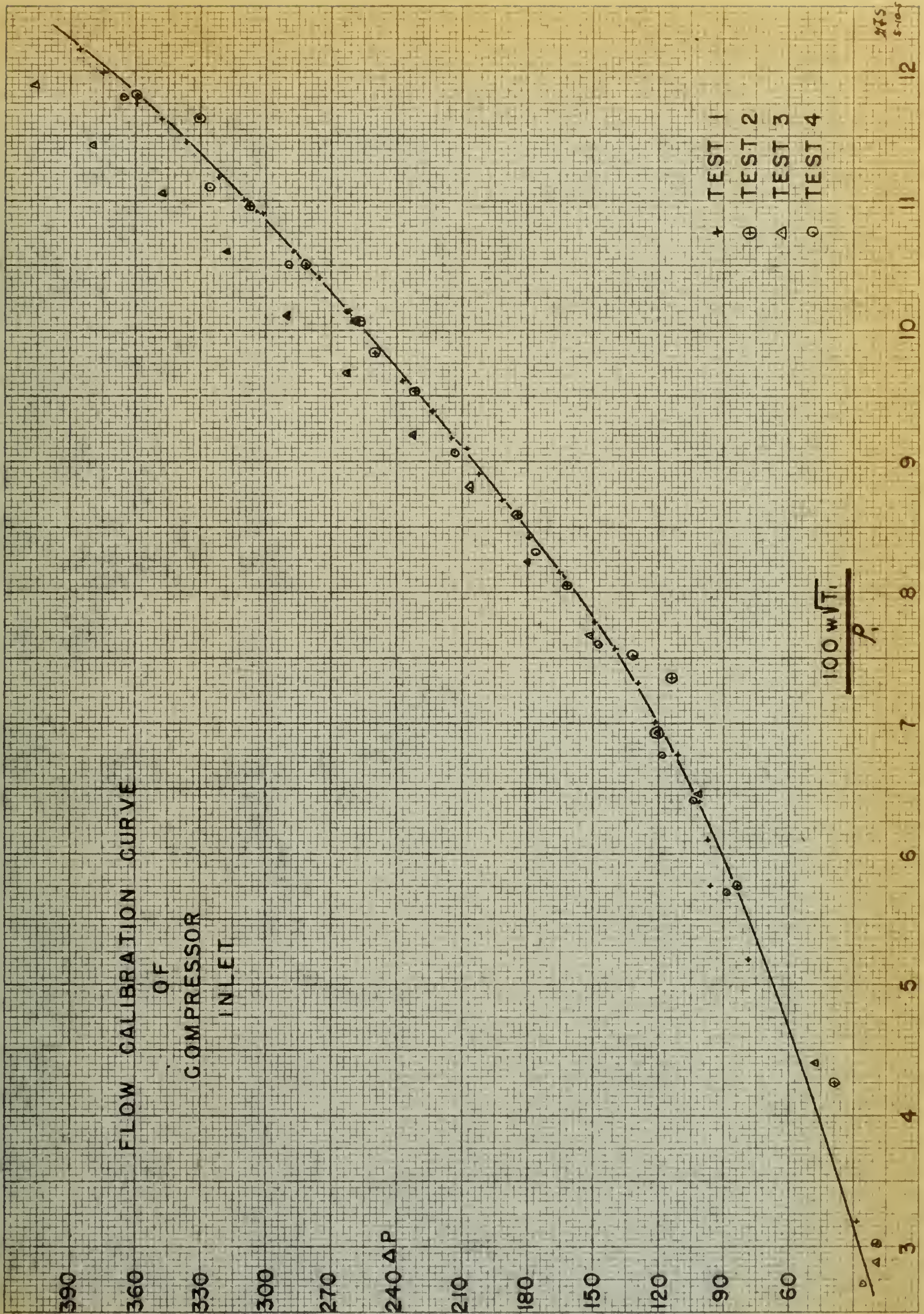

PARTIAL
COMPRESSOR CHARACTERISTIC
$\frac{P_2}{P_1}$
1.12
1.10
1.08
1.06
1.04
$\frac{100\,w\sqrt{T_1}}{P_1}$
5
6
7
8
9
10
11
12
$\frac{N}{\sqrt{T_1}}$
650
620
590
520
420
#7.8
5-10-5

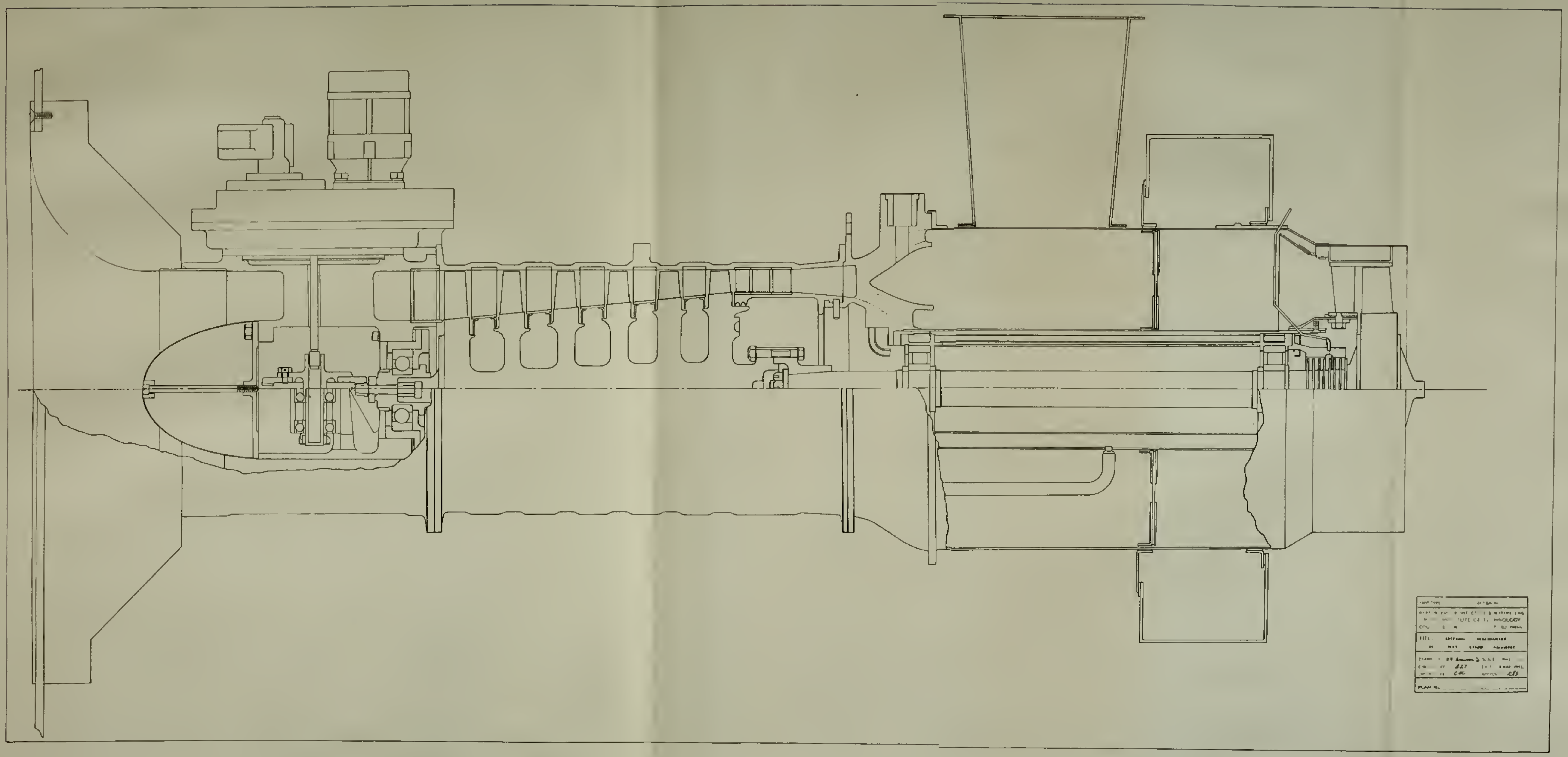
FIGURE X



INSTRUMENTATION DIAGRAM
FOR
COMPRESSOR TEST STAND

FIGURE XI


FLOW CALIBRATION CURVE
OF
COMPRESSOR
INLET
390
360
330
300
270
240 ΔP
210
180
150
120
90
60
3
4
5
6
7
8
9
10
11
12
$\frac{100\,w\sqrt{T_1}}{P_1}$
+ TEST 1
⊕ TEST 2
△ TEST 3
⊙ TEST 4
275
5-10-51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